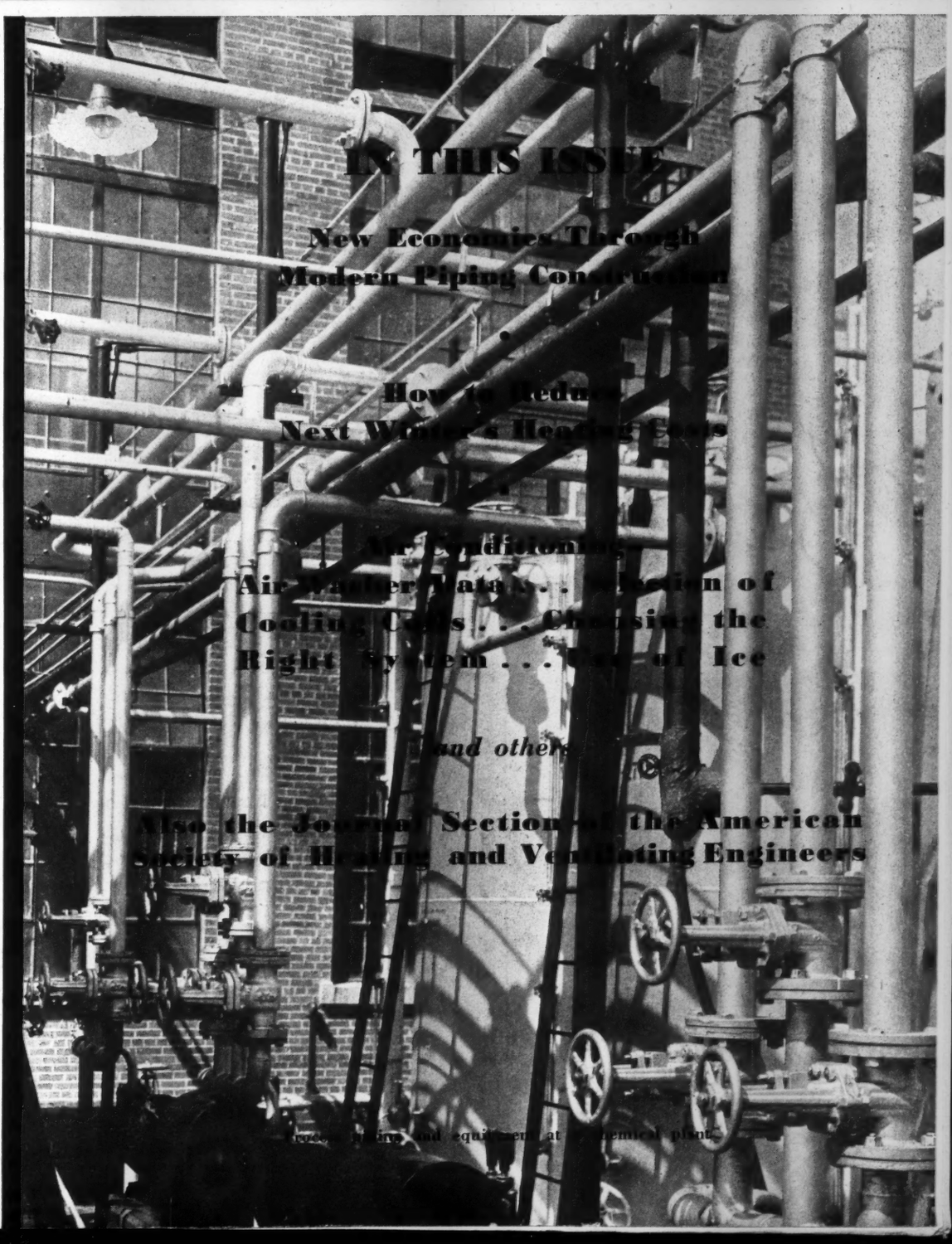
Table of Contents—page 25

Heating · Piping and Air Conditioning



IN THIS ISSUE

New Economies Through Modern Piping Construction

How to Reduce Next Winter's Heating Costs

Air Conditioning: Air Washer Details . . . Selection of Cooling Coils . . . Choosing the Right System . . . Use of Ice

and others

Also the Journal Section of the American Society of Heating and Ventilating Engineers

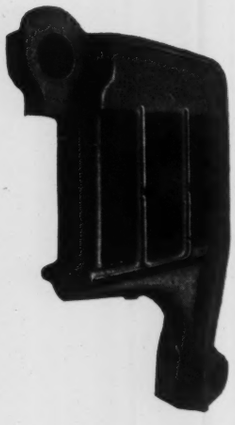
Power plant and equipment at a chemical plant

August, 1933

Heating · Piping
and Air Conditioning

Burnham Boilers

Sections are reversible. There are no rights and lefts. Every right is also a left. Every left is also a right.



How The Twin Section Burnham Cuts Down Installation Costs

. . . And Keeps Down Fuel Costs . . .

TAKE a replacement job for example. Suppose it replaces a steel boiler. If another steel is to be used, in most cases it means tearing out the building to take the old one out, and put the new one in.

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Burnham Boiler Corporation

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Representatives in All Principal Cities of the United States and Canada

THE EDITOR'S PAGE

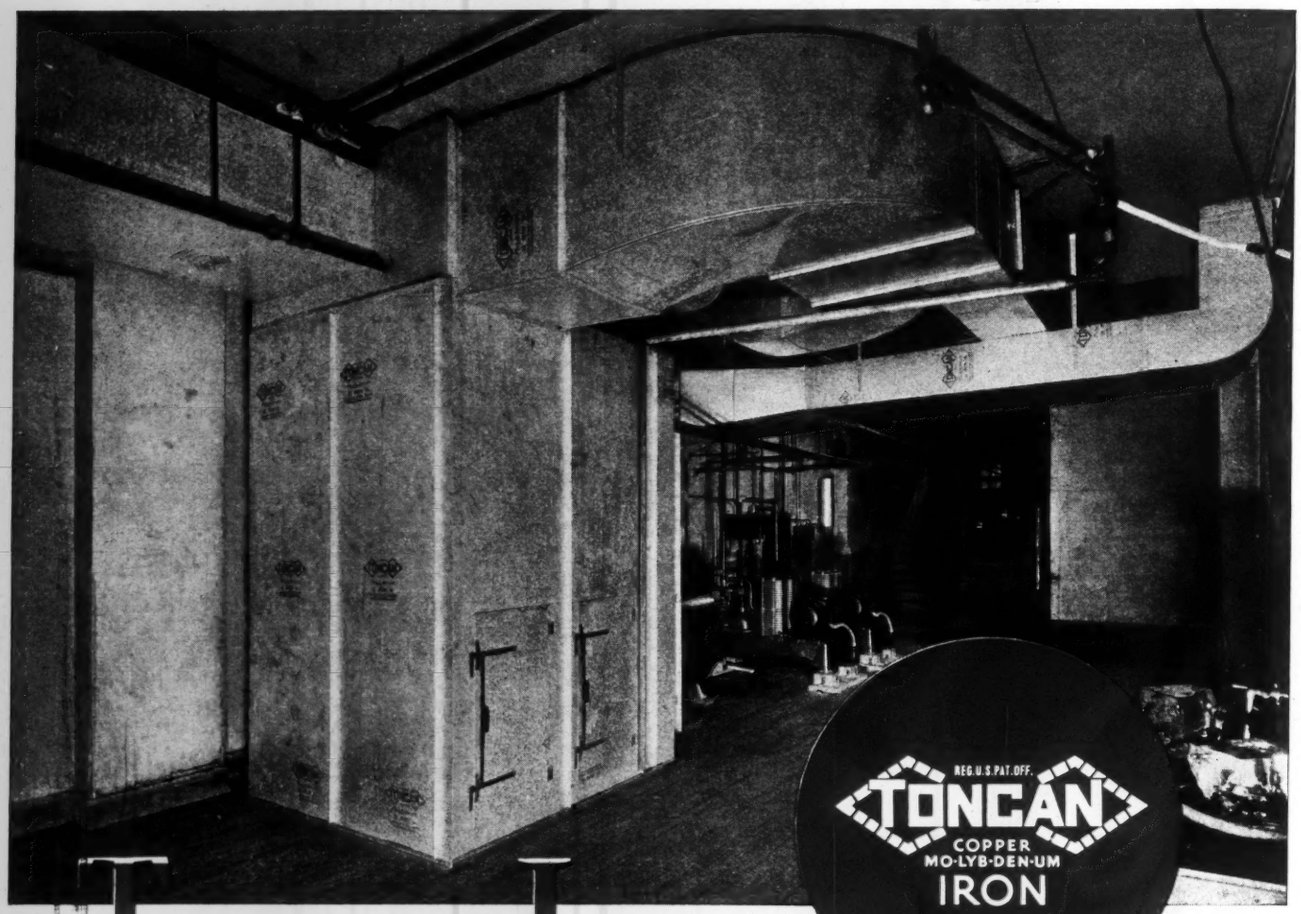
● This month—F. L. Snyder's articles on "Design of Modern Industrial Piping Systems," published in January and April, have been the subject of much favorable comment, having attracted wide interest from H. P. & A. C. readers; this month No. 3 of this title appears, discusses economies possible through the use of lighter wall thicknesses, describes actual examples . . . C. E. Baker's article tells how air conditioning is applied to maintain the quality of fruits and vegetables, another application of this much-talked about art . . . Under the heading "Practical Heating," Harold J. Taylor draws from his wide experience to point out what not to forget in enlarging boiler capacity or replacing old boilers—a timely contribution on modernization . . . Applications of lead piping for process lines is the subject of a short article starting on page 373 . . . A. Warren Canney's article—part of a series addressed to prospective owners of comfort air conditioning systems—is devoted to the central station type of job this month . . . How to reduce next winter's heating costs deserves consideration now, was ably covered by J. Earl Seiter at the recent meeting of the building owners and managers; his paper appears in this issue. Better heating is more economical heating, consequently deserves prime consideration when plant or building modernization is up; frequently, little or no investment is needed to achieve some measure of savings, some improvement in results . . . William Goodman shows how to solve problems involving two stage air washers with a chart developed by him; his article includes examples . . . Supplementing his discussion of the variable items in comfort cooling design published last month, Louis A. Harding presents in this issue a cooling coil selection chart . . . A report on the cost and consumption of ice for air conditioning, by Clifford F. Holske, is briefed this month . . . The front cover shows process piping and equipment at a chemical plant; photo courtesy The Linde Air Products Co.

● The *Journal Section* of the *A. S. H. V. E.* includes the Proceedings of the 1935 Semi-Annual Meeting, and two of the papers presented there:—"A Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads," by F. H. Faust, L. Levine, and F. O. Urban; and "An All Electric Heating Cooling, and Air Conditioning System," by Philip Sporn and D. W. McLenegan.

● From Shanghai comes a letter commending Erwin L. Weber's article on hot water heating in last May's issue as "the most sensible I've read in a long time"; previous comments have been received from readers in many parts of the U. S. and from Europe. The article described the importance of equalizing hot water heating systems, and Mr. Weber told how he balances flow through each unit of radiation to reduce fuel waste, promote heating efficiency . . . Heating wallpaper is produced by an English manufacturer, says the International Nickel Co., which is composed of a resistance metal of very fine nickel copper wire, put into the walls of a room, and connected to the electrical supply. Under thermostatic control, it can be heated to any desired temperature up to 85 F.

● Preliminary data on the number of air conditioning installations made in various cities during the first six months of 1935 indicate substantial gains as compared with the corresponding period of last year . . . J. M. Fernald, Baker Ice Machine Co., told Nebraska hotel men at their recent meeting that he knew of no business enterprise that requires air conditioning to the extent that hotels do, because hotels are a 24-hour-a-day institution giving 24-hour-a-day service . . . John Howatt predicted to the Chicago Real Estate Board last month, during a talk on air conditioning, that some one of these days health ordinances will enforce air conditioning for the average residential building, just as ordinances now require a certain amount of heating during winter months; already, said Mr. Howatt, air conditioning is a practical necessity in all commercial and better class residential buildings . . . Best week on record for output of electricity occurred last month, according to Edison Electric Institute figures; increased use of electricity for summer air conditioning and cooling, and the step-up in manufacturing activity getting the credit; output was 1,807,037,000 kw-hr . . . Talking about big figures, $2,255,000 was added to the deficiency appropriation bill by the Senate appropriations committee to provide for air conditioning the entire Capitol, the Senate and House office buildings; probably one of the effects of Congress' summer session . . . The contract awarded to York Ice Machinery Corp. several weeks ago for air conditioning the entire Interior Department building (except the top seventh floor, air conditioned last year) is said to be the largest single air conditioning contract ever placed, amounts to approximately $1,000,000. Capacity will be 1,000 tons, 500,000 cfm . . . Demand for air conditioning in Canada is likely to increase, according to Consul Damon D. Woods, Toronto, in a report made public by the Commerce Department, with humidification and purification rather than cooling stressed because of the long, cold winters and relatively short summers . . . "The effect of air conditioning on our business has already been quite gratifying, with this month's business substantially ahead of July, 1934," said Edward C. Blum, president of the Abraham & Straus store, Brooklyn, N. Y., during celebration of the completion of the store's new 1050 ton, 405,000 cfm system which supplies 90 per cent of the selling space.

● The nation's principal manufacturing industries have now been forced into the market for construction and machinery, in order to meet production requirements for their own goods, George A. Bryant, Jr., of the Austin Co., engineers and industrial builders, said last month. "Proof of this fact is to be found in expenditures involving more than $10,000,000 for expansion, modernization and equipment of plants where we are now engaged in active building construction," Mr. Bryant said. "The first half of 1935 yielded us 100 per cent more industrial construction business than we had in the first six months of 1934, which already represented two years of recovery from the low of 1932. This is an impressive advance, which bids fair to continue."



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CONTENTS *for* AUGUST, 1935

Vol. 7, No. 8 • Pp. 365-418

Design of Modern Industrial Piping Systems—New Economies Through Modern Piping Construction, by F. L. Snyder.. 365

Air Conditioning Maintains Quality of Fruits and Vegetables, by Clarence E. Baker..... 369

Practical Heating, by Harold J. Taylor..... 372

Lead Piping for Process Liquids..... 373

Choosing the Right Air Conditioning System, by A. Warren Canney 375

How to Reduce Next Winter's Heating Costs, by J. Earl Seiter 378

Two Stage Air Washer Problems Solved by Chart, by William Goodman 383

Selection Chart for Cooling Coils, by Louis A. Harding..... 386

Air Conditioning with Ice, by Clifford F. Holske..... 387

Equipment Developments (back section)..... 82

Book Reviews (back section)..... 87

Booklets, Reports and Papers (back section)..... 88

Conventions and Expositions (back section)..... 88

Recent Trade Literature (back section)..... 90

Index to Advertisers (back section)..... 96

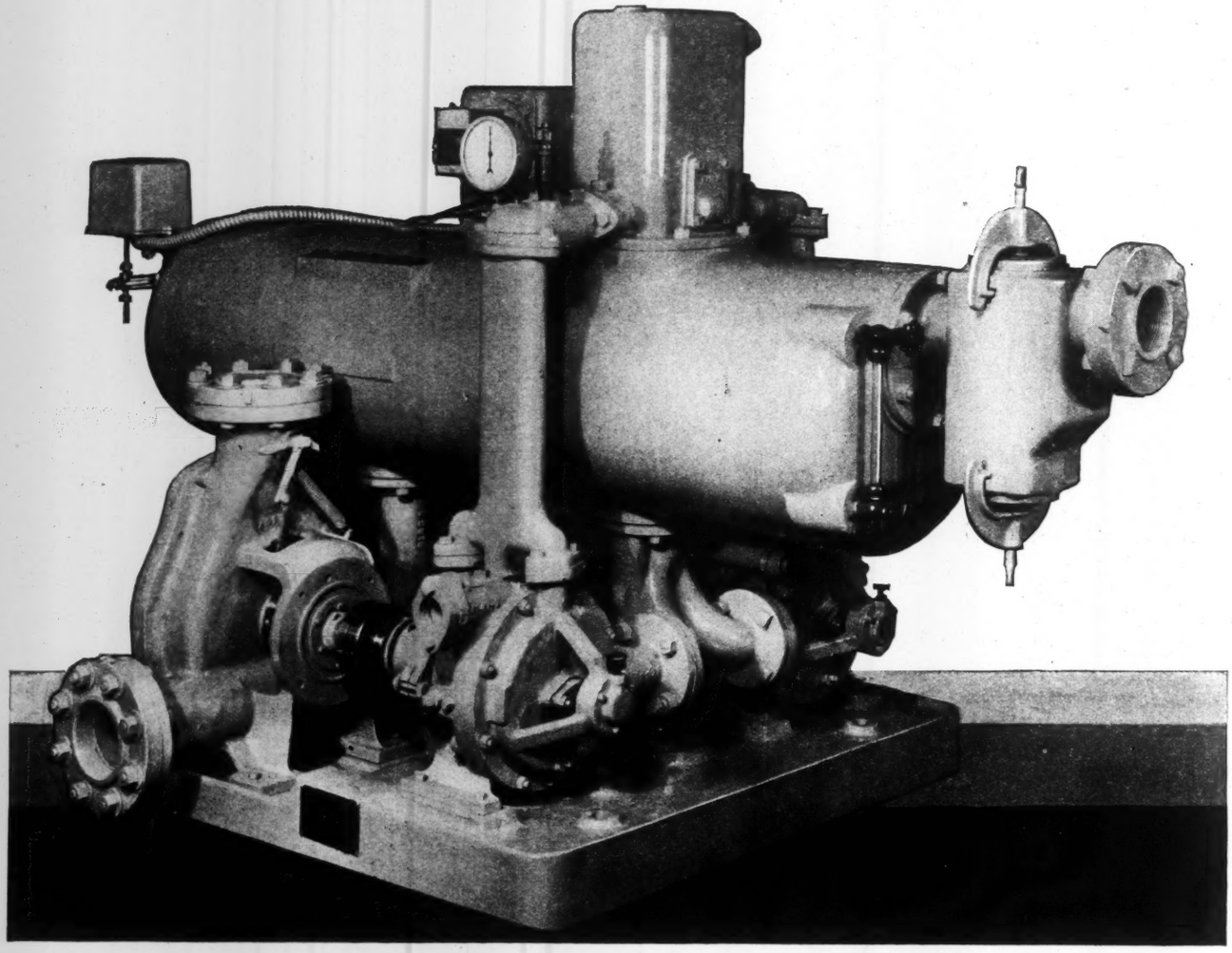
Journal Section of the American Society of Heating and Ventilating Engineers

A Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads, by F. H. Faust, L. Levine and F. O. Urban..... 391

An All Electric Heating, Cooling and Air Conditioning System, by Philip Sporn and D. W. McLenegan..... 402

Proceedings of the Semi-Annual Meeting 1935..... 410

Bulletin Section 416



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Heating · Piping and Air Conditioning

Vol. 7, No. 8 August, 1935

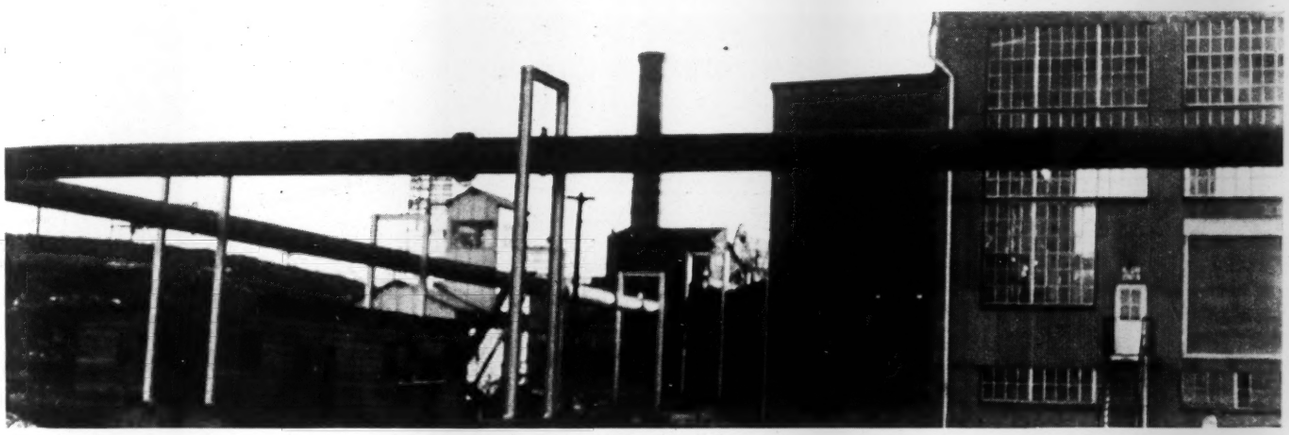


Fig. 1—Steam line with outside diameter of 24 in. and wall thickness of 0.1875 in. for saturated steam at 30 lb. per sq. in. gage

DESIGN OF MODERN INDUSTRIAL PIPING SYSTEMS

By F. L. Snyder*

NEW ECONOMIES THROUGH MODERN PIPING CONSTRUCTION

CROSS-COUNTRY gas and oil lines illustrate the advantages gained when pipe wall thickness is in proportion to the pressure and temperature of the fluid being transported. It is doubtful if these lines would have been justified if only the existing commercial standard wall thickness pipe had been considered. In these cases, the pipe wall was designed to provide for the pressure and temperature, with a reasonable safety factor. No consideration was given a mechanical safety factor for the welded joints, since they permitted the full thickness of the pipe wall to be maintained throughout the entire length of the line.

Design and construction of most industrial plant welded piping systems conform with commercial wall thickness standards established when screw coupling or screw flange type joints were practically the only known methods of pipe joint construction. In these standards, thickness of the pipe wall is governed not only by the internal pressure in the pipe line but also by the amount of metal removed by the threading die. This varies from about 20 to 50 per cent with the nominal diameter of the pipe and/or standard or extra heavy wall thickness. This necessitates a mechanical strength factor which materially increases the pipe wall thickness. In addition to this, corrosion is also considered and additional metal is added to prolong the life-span of the pipe.

Thin Wall Piping Standard Advocated

Today we have different and better materials of construction, and the method of joining pipe ends has radically changed since the original pipe standards were conceived. With the introduction of the welded joint, the flanged lap joint, the flanged expanded joint, and the new capillary attraction fitted joint, mechanical safety factors may be disregarded so far as the pipe joint is concerned, and the pipe wall thickness reduced accordingly. It is for this reason the author advocates an additional pipe wall thickness standard in order that new economies may be gained through modern pipe joint construction.

Welding fitting and valve manufacturers are developing standard products for light wall piping systems. In fact, existing installations with valves welded into the

*Engineering Department, E. I. duPont de Nemours & Co., Inc., Wilmington, Del.

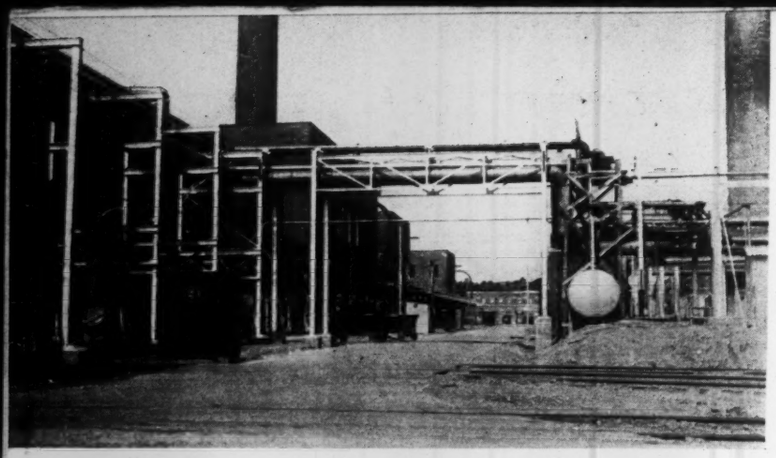


Fig. 2 — Twenty-inch outside diameter steam line of similar design to that shown in Fig. 1

●The author advocates the use of lighter wall thicknesses in order to obtain economies in the cost of industrial piping systems, explains why and how this may be accomplished, describes and illustrates typical jobs to prove his point.

line have been operating successfully during the past few years. The demand for valves with connecting joints prepared for welding is rapidly increasing, and some manufacturers are carrying them as a stock item. One manufacturer advises he has prepared valves for welding into systems having working pressures as high as 900 lb per sq in. and temperatures up to 800 F.

Table 1 shows the percentage of metal removed from the pipe wall by the threading die. It is based on the American Standard (Briggs) pipe thread, and it is assumed that all threads are perfect. In addition, the table shows the pressure permitted by the formula of the *A. S. M. E.* pressure piping code committee, when threaded joint construction is used as well as welded joint construction. The pressures are based on a fibre stress of 7000 lb per sq in., which allows a pipe wall safety factor of 6.5. It will be noted that the percentage of metal removed in the case of the threaded joint does not correspond to the percentage of increased working pressure allowed where the welded joint is used. In preparing the welded joint the full wall thickness is maintained; in fact it is strengthened by additional metal thickness in the completed joint.

From Table 1 it can readily be seen how much of the effective wall section is removed to make the connecting joint where screw type construction is used. The fact that the full wall thickness of the pipe is maintained throughout when joining the ends by welding should promote the use of a thinner wall pipe. Some manufacturers are producing commercial size pipe and welding fittings in O.D. sizes, lap or electric welded seams, up to about 24 in., with wall thickness as low as 0.1875 in., but in the nominal diameter sizes, without special order, nothing with the exception of tube can be obtained with less than 0.25 in. thick walls, and this only in the 8, 10, and 12 in. sizes. In many cases it is impossible to obtain these sizes in small quantities, probably because they are not considered standard items.

According to the *A. S. T. M.* piping specifications A-106, pressure not exceeding 250 lb per sq in. and temperature not exceeding 750 F is permissible in standard full weight wall thickness lap-welded pipe up to and including 10 in. diameter, and for the 12 in. nominal diameter, pipe wall thickness shall be 0.406 in. In extra heavy wall thickness pipe 400 lb per sq in. is permissible with the same working temperature in all nominal diameter sizes. Since the average pressure in industrial plants, especially in the oil and chemical industries, is about 100 to 150 lb per sq in., and in many cases below 50 lb, and since the conventional method of joining pipe is by welding, there is no doubt that a standard commercial pipe, having a wall thickness lighter than now available should be obtainable. The wall thickness of this pipe could be approximately those thicknesses shown in Table 2 under the heading "Permissible Wall Thickness, 150 lb per sq in. Working Pressure." This would not only insure economies in the pipe but also in the labor and materials required to complete construction.

The *A. S. M. E.* pressure piping code committee, has tentatively adopted the following formula for determining the minimum thickness of pipe walls, and for temperatures not exceeding 750 F. This section of the code reads as follows:

THICKNESS OF PIPE

Power Piping Systems (a)

In determining the minimum thickness on inspection to be used at different pressures and for temperatures not exceeding 750 F for steel or wrought iron pipe; 406 F for brass and copper pipe; and 450 F for cast iron, Formula I shall be used.

$$t_m = (PD/2S) + C \text{ or } P = (2S/D)\ (t - C)$$

where

t_m = Minimum pipe wall thickness in inches, allowable on inspection.
P = Maximum internal service pressure in lb per sq in. gauge (plus water hammer allowance in case of cast iron pipe conveying liquids, see Par. 122d).
D = Actual outside diameter of pipe in inches.
S = Allowable stress in material in lb per sq in.
C = Factor for allowance for threading, mechanical strength and/or corrosion.

(b) The value of C used in the Formula I shall not be less than that given for the respective material in Table I.

Table I—*C* Values

MATERIAL	VALUE OF *C* IN INCHES[a]
Cast iron pipe, centrifugally cast or cast horizontally in green molds	0.14
Cast iron pipe pit cast	0.18
Threaded steel, wrought iron, or non-ferrous pipe	Depth of thread, in.
Grooved steel, wrought iron or non-ferrous pipe	Depth of thread, in.
Plain end[b] steel, wrought iron or non-ferrous pipe, for 1 in. size and smaller	0.05
Plain end[b] steel, wrought iron or non-ferrous pipe, for sizes above 1 in.	0.065

[a] The values of *C* stipulated above are such that the actual stress in the wall of the pipe is less than the values of *S* set out below as applicable in Formula I.

[b] Plain end includes pipe joined by flared compression couplings, laps or van stone joints, and by welding, viz. *by any method which does not reduce the wall thickness of the pipe at the joint.*

(c) The value of S in Formula I shall not exeeed that given in Table II for the respective material:

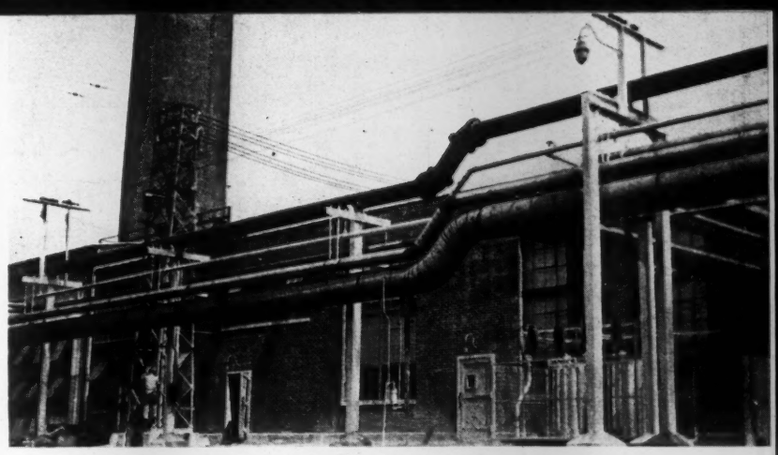


Fig. 3—Another view of the line shown in Fig. 2. The 14 in. water line at the top of the supports is rubber lined. This line is used as a continuous beam between supports, adding to the strength of the supporting structure

Table II—*S* Values

MATERIAL	VALUES OF *S*, LB PER SQ IN.
Steel pipe seamless, Grade C (A.S.T.M. A-106)	14,000
Steel pipe seamless, Grade B (A.S.T.M. A-106 and A-53)	11,500
Steel pipe seamless, Grade A (A.S.T.M. A-106 and A-53)	9,000
Steel pipe lapwelded (A.S.T.M. A-53)	7,000
Wrought iron pipe, lapwelded (A.S.T.M. A-72)	5,300
Wrought iron pipe, buttwelded (A.S.T.M. A-72)	4,500
Brass pipe, seamless (A.S.T.M. B-43)	4,500
Copper pipe, seamless (A.S.T.M. B-42)	4,000
Copper tubing, seamless (A.S.T.M. B-75)	4,000
Copper tubing, seamless (A.S.T.M. B-88)	4,000
Cast iron pipe centrifugally cast or horizontally cast in green molds (F.S.B. WW-P421)	6,000
Cast iron pipe pit cast (A.W.W.A.)	4,000

Referring to Table 1, the pressures shown are based on the formula of the *A. S. M. E.* pressure piping code, $P = (2S / D)(t - C)$. In the case of the threaded joint, the value of the constant C is the thread depth with no allowance for a mechanical or corrosion safety factor, while the value of C for the welded joint is 0.05 for pipe 1 in. and smaller, and 0.065 for pipe larger than 1 in. It is true that the exposed threads on a completed threaded joint are not quite so deep as the perfect thread, but the difference is slight, not nearly so much as the penalty given the welded joint. In fact, the effective thickness of the pipe walls in the case of the threaded joint is no greater than the thickness of the wall less the depth of the imperfect thread. Therefore, to balance conditions between the threaded and welded joint a mechanical and corrosion safety factor should be added to the value of C in the case of the threaded joint.

If the pipe walls were designed with consideration given the strength of the structure in the same way as with any other structure (for instance, the steel structure of a building), the only factor that would be pertinent is that of safety. Corrosion is always an indefinite factor. If corrosion enters the problem of design, it would in all probability be more economical to use corrosion resisting materials, or steel pipe lined with corrosion resisting materials, the strength of which should be in proportion to the pressure and temperature of the fluid, plus a reasonable strength safety factor.

Table 2 has been prepared to show the allowable pressures in standard commercial full weight wall thickness pipe in nominal diameters from 1 in. and up to 12 in. in accordance with the code. It is based on a permissible factor C value of 0.05 for pipe 1 in. and smaller, and 0.065 for pipe larger than 1 in.; the fibre stress used is 7000 lb per sq in. The C value considers the mechanical strength and corrosion of a welded joint, as interpreted by the code committee (see paragraph [a] of the code). This table also shows the wall thickness required if the pressure does not exceed 150 lb per sq in., and is based on the same factors. It is apparent that the wall thickness of materials used in most piping systems is not in proportion to the pressure and temperature of the fluid being transported. Therefore, in order to obtain a good yet economical design, the thickness of the pipe wall should be considered and the system designed accordingly. In many cases it will be found thin wall pipe can be manufactured from plate, the longitudinal seams of which are welded, at a lower cost than if commercial wall thickness pipe were used. Of course, certain requirements are necessary, among which is the qualification of the fabricator and the welder.

A properly designed piping system should be based on the characteristics of the fluid to be transported, such as the pressure, corrosive or erosive action on the pipe walls, temperature, and the type of the joint. There are many good alloys that can be used where excessive corrosion is experienced which is preferable, the author believes, to the use of extra strong or double extra strong pipe walls in order to increase the life span of the piping system against corrosion.

Some recent installations have been fabricated

Table 1—Percentage of Metal Removed from Pipe Wall by Threading

Nominal Diameter, Inches	Commercial Pipe Wall Thickness, In.	Thread Depth, Inches	% Metal Removed	Permissible Working Press., Threaded Joint, Lb per Sq In.	Permissible Working Press., Welded Joint, Lb per Sq In.	% Press. Diff.
1	0.133	0.0696	52.3	675	885	23.7
1¼	0.140	0.0696	49.7	593	633	6.3
1½	0.145	0.0696	48.0	555	589	5.7
2	0.154	0.0696	45.2	497	525	5.3
2½	0.203	0.1	49.0	501	672	25.4
3	0.216	0.1	46.3	463	604	23.4
4	0.237	0.1	42.2	426	535	20.4
5	0.258	0.1	38.8	397	485	18.2
6	0.280	0.1	35.7	380	455	16.5
8	0.322	0.1	31.1	360	417	13.7
10	0.365	0.1	27.4	345	391	11.8
12	0.375	0.1	26.6	302	340	11.2

Table 2—Present Standard and Proposed Wall Thickness

Nominal Diameter, Inches	Commercial Pipe Wall Thickness, In.	Permissible Working Press. Piping Code, Lb per Sq In.	Permissible Wall Thickness 150 Lb Work. Press. Piping Code	% Diff. in Metal Thickness
1	0.133	885	0.0641	51.8
1¼	0.140	633	0.083	40.7
1½	0.145	589	0.085	41.1
2	0.154	525	0.090	41.6
2½	0.203	672	0.096	52.7
3	0.216	604	0.103	52.3
4	0.237	535	0.113	52.3
5	0.258	485	0.125	51.5
6	0.280	455	0.136	51.4
8	0.322	417	0.157	51.3
10	0.365	391	0.180	50.7
12	0.375	340	0.202	46.2

throughout when the desired wall thickness was not available in commercial sizes, at a substantial saving in time and money, the pipe being rolled from plates with both longitudinal and circumferential seams welded. The thinner wall pipe also increased the flexibility of the structure, which made possible the elimination of expansion joints or bends that would have ordinarily been used to care for thermal expansion if standard wall thickness pipe had been used. This also reduced materially the size of anchors, hangers, and supports and the time required to complete each field joint, and incidentally the material required for making the joint.

Examples Indicate Possible Economies

As an example, consider a 10 in. pipe line transporting steam at a working pressure of 100 lb per sq in. According to the commercial pipe standards, and in fact general practice, this duty would require a pipe with a wall thickness of 0.365 in., while the actual allowable thickness according to the formula of the power piping code (providing the joints are welded) would be a pipe with a wall thickness of 0.218 in., based on a fibre stress of 7,000 lb per sq in., which represents a safety factor of 6.5 (from the physical properties of lap-welded open hearth pipe *A. S. T. M. Spec.* A-106) or approximately 40 per cent lighter than the present standard commercial full weight pipe. In invested capital this amounts to a considerable sum. Assume that the pipe line is 1,000 ft long, the weight of the pipe having a wall thickness of 0.365 in. is 40,483 lb while the weight of the pipe with a wall thickness of 0.218 in. is 24,290 lb, a difference of 16,193 lb. The unit price of pipe at this time is about $0.04 per lb. By using the thin wall pipe a saving of $647.72 is made possible on an apparent investment of $2,428.98, or approximately 26 per cent. This saving does not consider the reduced cost of the welding rod, gas, hangers, anchors, expansion joints, or labor, which could be safely estimated at 10 to 15 per cent additional.

The substitution of light weight pipe for commercial full weight pipe will lead to other economies in addition to those described; it can be fabricated and shipped in lengths 40 ft long, without the penalty of premium price. Therefore, in a pipe line 1,000 ft long, 25 field joints will be required, whereas, in commercial standard pipe, considering average lengths of 22 ft, 46 joints will be required.

In Germany, as well as other European countries, this condition has been recognized, since welded pipe joints are more or less in universal use. The commercial pipe standards have been governed accordingly. In Table 3, American and German commercial pipe wall thicknesses are compared. The American standard is in accordance with *A. S. T. M. Spec.* A-106,

Table 3—American and German Standard Wall Thicknesses

Nominal Diameter, Inches	Commercial Pipe Wall Thickness "American," In.	Commercial Pipe Wall Thickness "German," In.	% Diff. in Wall Thickness
1	0.133	0.0984	26.0
1½	0.145	0.0984	32.1
2	0.154	0.1082	29.7
2½	0.203	0.1181	41.8
3	0.216	0.1279	40.8
4	0.237	0.1476	37.7
5	0.258	0.1575	38.9
6	0.280	0.1772	36.7
8	0.322	0.2559	20.5
10	0.365	0.2756	24.5
12	0.375	0.3150	16.0

which permits a working pressure not exceeding 250 lb per sq in., although the working temperature is somewhat higher, it being 700 F. The German standard (DIN2401 and DIN2449), permits a working pressure of 368 lb per sq in. for water, and 294 lb per sq in. for steam, and working temperature of 572 F. It will be noted that the German thicknesses are between 16 and 41 per cent lighter than the American standard, although threaded joints are still permissible, the difference being that the German thread depth is not so great as that of the American thread.

Fig. 1 shows a steam line having an outside diameter of 24 in., wall thickness of 0.1875 in., and length of about 950 ft. The steam is saturated, and the pressure is 30 lb per sq in. gage. If conventional minimum wall thickness pipe had been used, an expenditure of $2,707.50 would have been added to the installation costs, or a 100 per cent increase in the pipe cost. In addition, the theoretical welding time was reduced 1190 hr, the oxygen and acetylene 1130 cu ft, and the welding rod by 66 lb. The joints are 90 deg vee, butt type. The deflection of the line is cared for in the direction changes, which are made with welding fittings having the same wall thickness as that of the pipe. No expansion bends were necessary. (In the April, 1935, HEATING, PIPING AND AIR CONDITIONING the author discusses this method of caring for pipe deflection or expansion). The flanged joint shown in the foreground is used for cold springing, which in this case amounts to the full value of the theoretical thermal expansion, permitting the line, when hot, to return to the neutral position, it being in stress only when cold.

Similar economies were experienced in the installations shown in Figs. 2 and 3. In this case the steam line is 20 in. outside diameter. The steam pressure, wall thickness, and method of construction is the same as that for the line shown in Fig. 1. The 14 in. outside diameter water line shown on the top of the supports has a wall thickness of 0.1875 in. and is rubber lined. The joints are made with flanges, necessary due to the rubber lining, which would not permit field welded joints. The flanges were fabricated from plate and welded to the pipe before the rubber lining was applied. Ordinarily this type of structure would be placed underground, and cast iron bell and spigot pipe would be used. In this case, the overhead lines ran in the same direction as the water line and it was therefore found advantageous to put the line in the overhead position, since it not only reduced the cost of the installation but also added to the strength of the supports. The thermal expansion being negligible, the water line was used as a continuous beam between supports, thus adding to the strength of the supporting structure. The savings in this case were considerable as compared to a 14 in. cast iron underground line when all conditions were considered.

In another case where corrosive gas is transported through pipe lines, stainless steel was used in the fabrication of the pipe. The wall thickness was designed for the pressure and temperature without consideration of a corrosion factor. This also was found to be much more economical than ordinary piping having excess wall thickness, when the life spans of the two are compared. It was fabricated throughout, since commercial sizes in stainless steel pipe of the wall thickness desired were not obtainable.

Air Conditioning Maintains Quality of Fruits and Vegetables

Valencia oranges, California. Photo courtesy California Fruit Growers Exchange

By **Clarence E. Baker***

● **How air conditioning is applied to maintain the quality of fruits and vegetables is described here, and results of recent research in this field are surveyed briefly.....The various methods of precooling, cooling in transit, and treatment with carbon dioxide and other gases are among the subjects discussed.**

DEVELOPMENT of improved varieties of fruits and vegetables possessing superior quality is of little value to the ultimate consumer if the produce reaches him only after this high quality has been lost through careless or improper methods of handling after harvest; consequently, much attention is given to the proper handling of quickly perishable produce.

An article in the January, 1932, H. P. & A. C. emphasized the importance of rapidly reducing the temperature of quickly perishable fruits and vegetables to check their ripening and the resulting loss of keeping and eating quality. It was pointed out that even when the produce was loaded as quickly as possible into iced refrigeration cars, it was difficult to cool the load rapidly and evenly to a temperature sufficiently low to check the ripening process going on within the fruit, to a degree that it can be safely handled. These conditions result in the rapid and uneven ripening of the produce so that there is likely to be a considerable loss of quality, due to overripeness or actual spoilage, before the produce is marketed or consumed.

The importance of proper arrangement of the load to permit the maximum circulation of air about the packages, the use of floor racks to aid air circulation within the car, and the use of salt upon the ice further to reduce the air temperature was stressed. The necessity of a positive circulation of air to accomplish the required amount of cooling in the time available for the process has lead to the development of several types of portable precooling outfits that were briefly described. In general these devices are electrically driven blowers to cause rapid movement of air through an ice and salt mixture in the refrigerator car ice bunker, so that the load is cooled quickly and evenly. After this initial cooling the car is re-iced and closed.

Car precooling is also accomplished by moving the loaded car to a central cooling plant where a stream of refrigerated air is passed through the ventilating system of the car by means of flexible ducts. Sometimes entire trains of cars are cooled simultaneously by connecting the cars through the hatches by collapsible ducts.

The warehouse system of precooling used in some large fruit shipping centers also was described. In this system the produce is not loaded directly in cars from the field, but is first placed in a refrigerated room at a low temperature until the field heat is removed. The product is then loaded into iced refrigerator cars and started on its way to market.

Cooling Apples in Packages

In a recent publication[1] Lloyd and Decker report the results of comprehensive studies of the factors involved in the cooling of apples in packages. Grimes apples packed in wood veneer tub bushel baskets, with ventilated paper liners and corrugated paper facing pads, were placed in a specially designed refrigerated tunnel in which both temperature and air velocity could be controlled. Cooling of the apples in all parts of the package was studied by means of thermocouples placed at various locations within and outside the package.

During the early part of the cooling period the outer rows of apples cooled much more rapidly than the inner rows; this temperature gradient became less after several hours and as cooling progressed the temperatures throughout the package tended to become uniform. Value of air spaces between the fruits in the package was demonstrated by filling the spaces with mineral wool. Fruit in packages so treated cooled much more slowly, indicating that the air spaces permit a more rapid change of temperature, in spite of the slow circulation of air taking place within the package under low air velocities.

The effect of the velocity of the air stream on the rate of cooling was investigated and found to be very important. Velocities below 100 fpm did not affect the rate of air movement in a lined bushel basket, but an air velocity of 156 fpm increased the circulation of air between the apples and consequently the rate of cooling.

*Purdue University Agricultural Experiment Station.

[1]Lloyd, J. W. and Decker, S. W. Factors influencing the refrigeration of packages of apples. Ill. Agr. Expr. Sta. *Bul.* 410.

Size of the fruit had no detectable effect upon the cooling rate so long as the total weights of the content of the packages were similar.

This type of package was found greatly to change the cooling of apples under refrigeration. Fruit packed in ventilated corrugated paper bushel boxes cooled as rapidly as lined tub bushel baskets, but unventilated corrugated bushel boxes cooled much more slowly. Western apple boxes retarded the cooling of the fruit much more than did lined veneer baskets. The presence of a ventilated paper liner in the bushel basket retarded the rate of cooling but little as compared to the unlined basket.

Fruit wrapped in oiled paper cooled much more slowly than unwrapped fruit. The writers report that "under conditions found in the average refrigerator car, it would probably require two days longer to reduce the temperature of a carload of wrapped fruit to the desired degree than a carload of unwrapped fruit, when shipped during the summer months."

This excellent piece of work answers many questions that have been raised concerning the behavior of fruit during cooling and it is of interest and value both from the standpoint of precooling and from that of refrigeration.

Advantages of Precooling

Intelligent precooling and careful subsequent handling makes possible a greater degree of ripening before harvest so that more nearly full maturity is obtained. Such produce has much more pleasing quality than that picked at a very immature stage and ripened after harvest, as is necessary when allowances must be made for uncertain shipping conditions.

The marketing of Florida strawberries in northern markets has presented many problems in handling and shipping, which are further increased by the necessity for washing the fruit, increasing its tendency to rot in transit. Precooling the fruit with chilled, circulating air has proved to be of great value by rapidly reducing the temperature of the fruit and also by removing the excess moisture, checking the tendency for rots to develop on the moist fruit. It has also been found that precooled fruit requires less ice in transportation than untreated fruit—an important item of saving on a long haul.

The development of portable precooling devices has made it possible for small groups or individual shippers to adopt this practice without large investments for mechanical equipment. Even at points where electricity is not available, precooling devices operated by gasoline power plants have been assembled on trucks or power take-off arrangements developed to operate blowers.

Several commercial companies in different sections of the country have been organized for the purpose of selling air conditioning or precooling service. In most cases each company has developed and patented somewhat different types of air circulating and chilling devices, but the same general principle is involved. This service is very convenient for those to whom it is available, as individual shippers may secure the benefits of efficient precooling methods at reasonable cost. Some of these companies sell machines to individuals or shipping associations who prefer to do their own precooling. In addition to precooling, some of these concerns conduct prebleaching and preripening service, which is of great value in the case of certain crops.

Preheating for Cold Weather Protection

Another interesting phase of fruit and vegetable air conditioning is the preheating of loaded cars of such crops as apples and potatoes, which must be transported during excessively cold weather. The same type of equipment developed for precooling purposes may be modified to circulate warm air through refrigerator cars, heating the car and produce to such extent that long distance shipments can be made in severe weather without danger of freezing or chilling. Great caution must be exercised in using this as a method of thawing produce that has been frozen in transit, as frozen produce must be thawed at a temperature very close to the freezing point; rapid thawing sometimes tends to increase the injury to the produce. If properly thawed out, slightly frozen produce may suffer little injury.

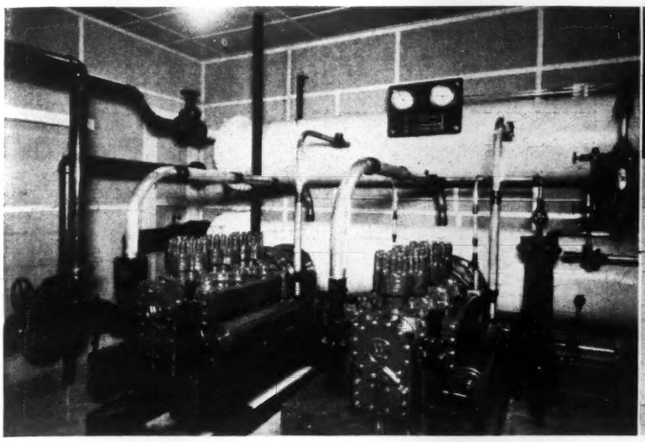
Precooling for World Markets

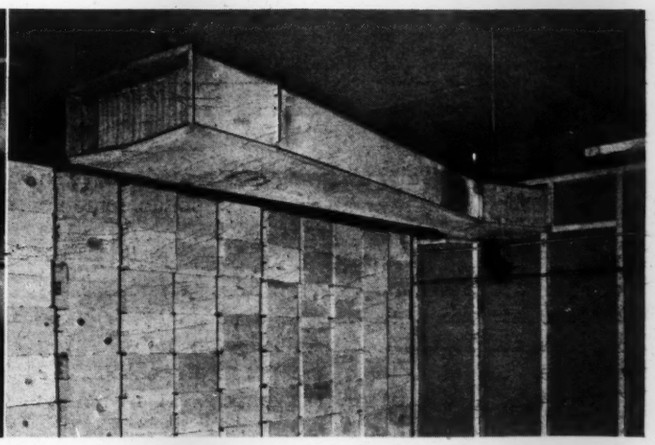
The importance of precooling fruit for transoceanic shipments is being realized more fully than before. Large precooling plants are being established at fruit shipping ports in many countries. During the last two years a plant having a capacity of 30,000 cases of citrus fruits at one time has been constructed at Lourenco Marques, Portugese East Africa. Loaded trucks are run directly into an air lock in the plant and unloaded on to small trolleys which are run into the refrigerated rooms. After the fruit is cooled the trolleys are moved to the loading platform and a crane carries the entire load to the hold of the ship in one trip.

One of the largest precooling plants in the United States has recently been constructed at a Florida port. The 600 ton refrigeration plant and air conditioner is located upon a pier and a battery of insulated barges equipped with refrigerator doors are used as floating precooling chambers. The barges, holding 3,000 cases each, are loaded with fruit, towed to the pier, and flexible air couplings are attached. The precooling air is circulated through the compartment about the fruit and returned to the air washer where it is recooled and conditioned and recirculated by means of powerful fans. The fruit is then loaded into refrigerated holds of the ships. Fruit temperature is reduced at the average rate of approximately 3½ F per hr, which is very rapid for wrapped fruit packed in boxes. To accomplish this it was necessary to use a sufficient pressure of air, together with a building up of a sufficient resistance to the air flow, to force the air through the boxes of packed fruit. This plant has the capacity to cool twelve barges simultaneously.

Dry Ice Refrigerator Cars

An American manufacturer has recently introduced a refrigerator car utilizing solid carbon dioxide or "dry ice" as the refrigerant. This car is heavily insulated and the produce is enclosed in a gas-tight shell outside of which the refrigerant is placed. This development, it is hoped, will accomplish what amounts to precooling during transit, thus saving an additional





This air conditioning plant at the La Verne (Calif.) Lemon Growers' Association was designed to maintain a constant temperature of 57 F and a relative humidity of 90 per cent. Installed by the Gay Engrg. Corp., it consists of two condensing units connected to a common condenser and flooded type evaporator.

The compressors start and stop automatically to maintain temperature and humidity conditions at the proper point. Because it is not necessary to run the condensing equipment during certain months of the year, and as the nights are usually cool and the humidity relatively high, the plant is interlocked with a fan control and outside air damper to take advantage of outside conditions in maintaining indoor conditions. All equipment, including the dampers, is under control of thermostats, humidistats, etc.

A special type of air conditioner, shown in the right view, was developed for this job, and a number of these units are suspended in various parts of the storage room, connected to an air supply duct. In the unit water from the refrigeration plant is sprayed through the nozzle which controls the temperature of the air as well as humidity and also creates a natural air circulation through the unit. During some periods of operation it is not necessary to operate the fan for the circulation of air, inasmuch as the unit has a tendency to set up its own air circulation.

In the left view a vertical cylinder on the top of which there are two sight glasses is shown at right of the condensing units. This is an oil separator which separates the oil from the liquid freon, operating on the percolator principle.

Both condensing units are provided with spring supports and the discharge and suction lines are provided with flexible tubing. The oil equalizer connections between the two compressors are also provided with flexible tubing so that the compressors can float freely on the spring supports.

[Photos courtesy Westinghouse Electric & Manufacturing Company.]

24 to 36 hours in the distribution of rapidly perishable produce.

Carbon Dioxide Air Conditioning

Another interesting phase of air conditioning of perishable fruits and vegetables is the holding of them in accurately regulated and controlled quantities of carbon dioxide gas. An artificial atmosphere is created within the car or storage room by adding carbon dioxide until the desired concentration of the gas is obtained. This subject has been studied in great detail by Franklin Kidd and Cyril West of Cambridge University in England, and by workers in this country at the Boyce Thompson Institute for Plant Research and the U. S. Department of Agriculture.

The results of these combined investigations indicate that perishable fruit stored in increased concentrations of carbon dioxide keep as well at a considerably higher temperature than do fruits stored at their optimum storage temperature in a normal atmosphere. For example, with a 25 per cent concentration of the gas, the softening of warm peaches held at about 70 F has been checked equal to that accomplished by a drop in temperature of 18 F during the period following harvest. Sometimes both the carbon dioxide and oxygen content are regulated.

Attempts to substitute carbon dioxide treatment for precooling of peaches, strawberries, and other small fruits under transit, in tests conducted by the U. S. Department of Agriculture, have been successful both in checking the development of rots and reducing the rapidity of softening of the fruit. Concentrations of from 10 to 25 per cent of the gas were used. These treatments had an average effect of checking rots in the tops of iced refrigerator cars, where they are most serious, equivalent to lowering the temperature 21 F during the first 36 hours, or a reduction of 10 F for the entire trip. The fruit in the top of the treated cars was much firmer upon arrival at its destination than the fruit in the bottom of the control car shipped under ordinary refrigerated conditions, although the fruit in the bottom layers of the control car averaged about 13 F lower in temperature than similar fruit in the treated cars.

The unfavorable effect of carbon dioxide upon the flavor of certain products, however, sets definite limits to this method. Grapes, sweet corn, carrots, and peas appear to stand extreme concentrations of the gas without loss of flavor. Apricots, peaches, strawberries, and red raspberries appear to be easily injured by concentrations above 25 per cent, depending upon the temperature, as the injury for a given harmful concentration of gas increases with increases in temperature or the duration of exposure. Plums, apples, cherries, pears, oranges, and currants are among the fruits that show a considerable tolerance for higher concentrations of carbon dioxide. The first objectionable effect is a slight loss of aroma. With prolonged or severe treatments a loss of flavor occurs and in extreme cases a

PRACTICAL HEATING

By **Harold J. Taylor***

PARTICULARLY in heating plant modernization or remodeling work, many practical points come up which must be allowed for if the job is to be done with the minimum of trouble and expense. Last month, under the above heading, suggestions for installing radiators below the water line of a steam boiler were given; this month consideration is given to enlarging or replacing old boilers. These practical pointers are based on several years of experience with numerous remodeled heating jobs.

Enlarging Old Boilers for Added Capacity

When enlarging a boiler, be careful in checking size, style and number of boiler. The number plate on the boiler may read, for example, No. 1—S—8, but the boiler may have been added to previously and is at present a No. 1—S—10. As a safeguard, measure the width, length, and height of boiler and check with manufacturer's data.

Many heating contractors have enlarged round type boilers by adding a section or sections above the fire-pot. In nearly every case the boiler capacity was increased but a trifle by this procedure. If reference is made to any manufacturer's data books, it will be found that the adding of sections above the fire-pot does very little toward added capacity, but does add, as a rule (providing chimney is suitable to pull the additional flue resistance), better efficiency, especially on oil fired jobs.

In many cases it is more economical to furnish a new boiler of latest design where a larger boiler is required. Consideration should always be given this point where the boiler size is questioned.

Replacing Old Boilers

When required to replace an old boiler, determine the load by measuring the building for the proper amount of radiation required and not by the amount of radiation installed.

Be sure to determine the temperature desired inside for a specific temperature outside. This practice should be followed in every case.

Check the grate area and compare the two boilers in this respect.

Compare water line dimensions, smoke-pipe size, chimney size required, length, width and height of boilers.

Consider the header and flow connections.

If boiler is to be pitted, be sure that the sewer level is below bottom of pit.

If sanitary or rain sewers are encountered in the digging of pit, then these sewers will have to be re-routed to a point outside of the pit area.

Watch your step when confronted with the installation of a pit.

Be sure to figure for the wood form work, concrete work and disposal of dirt.

Be sure that the new boiler as selected will pass through openings, and that it will clear overhead obstructions, in connection with the construction of the building.

After following the above directions, then decide on make and size of boiler.

If the boiler is to be oil- or stoker-fired, consult the manufacturer furnishing the automatic heat equipment and advise him of the boiler selected. The boiler to accommodate his equipment may have to be provided with a pit or similar arrangement and it is best that these conditions be allowed for beforehand.

If the boiler being replaced has been mechanically fired, be sure the new boiler is adapted to the equipment, both in regard to type and size. If the new boiler is of larger capacity, it may be necessary to increase the capacity of the mechanical equipment.

Fruit and Vegetables—

(*Continued from preceding page*)

fermented odor develops and the fruit becomes insipid. Some varieties of apples, citrus fruits and other produce appear to be able to stand exposures to 50-60 per cent of carbon dioxide at temperatures up to 60 F for several days without injury to flavor. Under ordinary conditions of temperature in refrigerator cars an excessive or dangerous concentration of the gas probably is not necessary in order to accomplish the desired results. As refrigerator cars are not air tight the concentration of the gas generally falls during the course of the run through the entrance of fresh air. In tight storage compartments, however, the concentration of the gas tends to increase as the respiration of the fruit produces a considerable amount of carbon dioxide, which is given off into the air of the room.

This treatment is considered to be in the experimental stage and while the possibilities of its applications are wide, the details of satisfactory methods have not been worked out with sufficient certainty to make its use at present widely adaptable.

Other Gases for Air Conditioning

Other gas treatments for conditioning fruits and vegetables include the use of ethylene gas in the ripening and coloring of such crops as bananas, tomatoes, and Japanese persimmons. Reports have recently been published of the successful use of nitrogen trichloride in checking the development of mold spores on California navel oranges during shipment and storage. This process, also, is as yet in the experimental stage. These developments, however, serve to indicate some of the various types of air conditioning that show promise of becoming commercial practices in the handling of fruits and vegetables to permit the consumer to enjoy them in their prime condition.

*Contracting Engineer, Detroit, Mich.

Lead Piping for Process Liquids

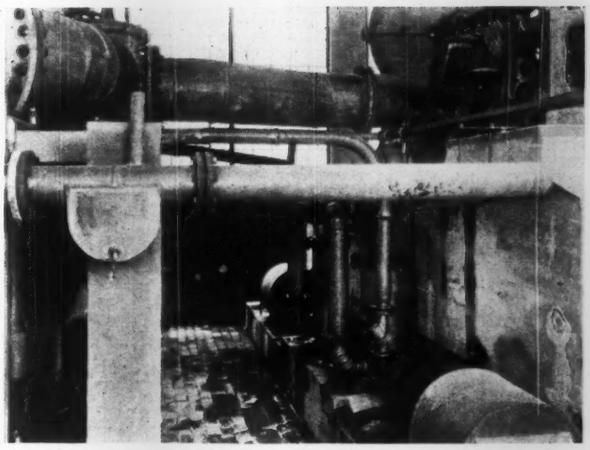


Fig. 1—Lead acid lines in a pulp and paper plant

A LARGE number of modern manufacturing industries require the use of chemicals corrosive to metals. Because of the high resistance of lead to many corrosive chemicals, particularly those commonly used, such as sulphuric acid, lead pipe is widely employed in handling and transporting them. Where no great strength is required, chemical or soft lead pipe is generally used. If greater strength and rigidity is needed, antimonial lead pipe may be employed. In a few cases, as handling salt water or hydrochloric acid, antimonial lead is more resistant to corrosion than soft lead. Where neither of these types of lead pipe have sufficient strength, lead-lined iron or steel pipe may be used.

A new alloy of lead with a small amount of tellurium has been developed and is now generally available. It has properties which indicate excellent possibilities, particularly for chemical work.

As a rough guide to the proper selection of lead pipe the following list of corrosive chemicals includes common corrosives with which lead is successfully used, but it by no means specifies all such chemicals, nor does it signify that lead is best under every condition.

Aluminum Sulphate	Reductions
Ammonia Vapor	Sodium Bisulphate
Ammonium Sulphate	Sodium Chloride Solution, or sea water
Antimony Tri-Chloride	Sodium Hydrosulphite
Bleach Liquors	Sodium Hyposulphite
Brown Acid Mother Liquor	Sodium Sulphate
Carbonates, soluble	Sodium Sulphide
Chlorine Gas	Sodium Sulphite
Hydrofluoric Acid	Sulphonations
Koch's Acid	Sulphur Chloride
Malachite Green Mother Liquor	Sulphur Dioxide
Mixed Acids	Sulphuric Acid
Nitration Mixture of H—Acid	Sulphurous Acid
Para-nitrophenol	Sulphuryl Chloride
Phosphoric Acid	Zinc Chloride
Phosphorous Chloride	

Temperatures used with lead pipe should be carefully watched. They should not exceed 550 F, in the case of "chemical lead" or 425 F, in the case of antimonial lead, and it is best to avoid fluctuating temperatures if possible. Local conditions may cause these figures to be raised or lowered somewhat.

As stated earlier, homogeneously lead-lined iron or steel pipes are often used. They provide greater strength and are advantageous for temperatures and pressures above those possible with chemical or antimonial lead pipe. Copper pipes lined with lead or lead alloys as a protection against the salt water they transport and lead and antimonial lead pipe are extensively used on ship board.

Possibility of alkaline corrosion of lead when it is in contact with concrete tanks, floors or other cement structures in which free lime may be present should be guarded against. In such cases the pipe should be wrapped in tarred paper or coated with a good bituminous coating, as a simple but effective precaution.

The flexibility of lead pipe often may be utilized in the fabrication of odd shapes. A good lead "burner" can readily make up heating coils or any other necessary apparatus on the job.

Valves, cocks and all necessary fittings for lead pipe lines are made of antimonial lead and have the same corrosion resistant characteristics as the pipe. Where lead-lined pipe is used, the fittings may be of the same metal lined with lead.

"Burning" or Welding Lead

As the chemicals handled in lead pipe are often corrosive to solder or are conveyed at temperatures high enough to soften the solder, joints in lead pipe are frequently made by "burning" or welding lead to lead (autogenous welding). This is accomplished by beveling the ends of the pipe and placing them together. They are also scraped clean with a shave-hook for about ½ in. back from the ends. The pipe is tacked together with a torch flame and the joint built up using burning bar and torch. The flame is applied to pipe and bar simultaneously and the joint built up slightly higher than the pipe. Temperatures employed should be between 625 and 700 F; higher temperatures will injure the lead.

Small, light torches should be employed and oxy-hydrogen, oxy-acetylene, oxy-calorene or oxy-illuminating gas flames. Small tips are preferable, the size increasing slightly with the thickness of the lead. The flame should be held approximately at right angles to the work, with the lead at about the tip of the inner cone. After the metals fuse together at one point the flame is flicked away to avoid burning the lead and is next applied at an adja-

This article is adapted from information made available through the courtesy of the *Lead Industries Association*.

cent spot. Lead welding requires skillful and rapid manipulation of the torch to avoid burning through the lead. For this reason the torch tip and valves are usually separated by a length of tubing. Thus the valves may be set down at a convenient distance from the work and only the tip itself need be moved, permitting much greater flexibility and delicacy of operation.

When the bottom of a pipe is inaccessible for "burning," the top may be split and turned back, permitting the bottom to be "burned" from the inside. After the lower half has been welded, the top is bent back into place, and ends and split "burned."

Many complicated pieces of apparatus can be fabricated quickly by the use of lead "burning."

Often lead pipe is made into coils for heating or cooling corrosives. Lead spacers are usually burned between

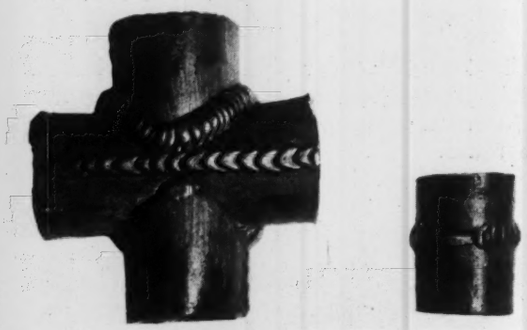


Fig. 2—Left, cross made of lead pipe burned. The bottom pipe has been tacked in place but not yet burned. Note how pipe has been scraped clean at joints. Right, simple burned joint in lead pipe partly made; note bevelling

each turn of these coils to support them. Another method is to use supporting legs of lead pipe larger in diameter than the coils, running from the bottom to the top. These pieces of pipe are split for their entire length and notched to fit each turn of the coils. The two pieces are then placed against the coil and burned together. In effect, the coil passes through a lead pipe leg at each of several points in its circumference. This method of support has the advantage that the weight of each turn is carried directly on the floor and not on the next lower coil. A tendency to crush the bottom coils which carry the weight of all those above them is thereby removed.

Heating coils of lead pipe are generally limited to 50 lb per sq in. steam pressure, whereas homogeneously lead-covered copper pipes are used for heating coils with steam pressures up to 125 lb per sq in.

Often it is desirable to fabricate large size lead pipe on the job by rolling the proper size of lead sheet around a mandrel and burning the longitudinal seam. Flanges may be fastened to lead pipe by cutting them from sheet lead and "burning" to the pipe.

Wiped Joints

The wiped solder joint is the oldest type of lead connection; properly made it has no superior. If a large number of joints are needed at one time, they can be made very economically by this method.

Wiping solder should contain from 38 to 42 per cent tin and the balance lead, although some prefer up to 2 per cent of antimony as well. Advantages are claimed for solders composed of lead, tin, and cadmium for wiping purposes because they have a wider temperature range of plasticity which gives them better workability, although they apparently show no superiority in the finished joints. A typical composition of this class of solder would be 67 per cent lead, 24 per cent tin and 9 per cent cadmium.

To wipe the ordinary joint connecting two pieces of lead pipe, flange out one end of the pipe so that its inside diameter becomes equal to the outside diameter of the rest of the pipe. Then shave down the outer surface until it is straight. Bevel the end of the other piece of pipe until it fits snugly within the first, giving a smooth bore inside and smooth surface outside. Cover each pipe about 4 in. back from the end with plumbers' soil or library paste. Scrape the pipe clean back from the ends a distance equal to one-half the length of the joint to be made and immediately apply tallow to the cleaned surface.

With the solder at a temperature of about 600 F, which will just char a pine stick or turn paper brown if dipped in it, pour a little solder on the top of the joint, holding the wiping cloth underneath. Move the ladle back and forth along the joint, pouring the solder on little by little until the pipe is heated, so that the solder will unite with it and until there is enough solder on the pipe and cloth to build up the joint. Then start forming the soldered joint, wiping from bottom to top and down to bottom again. Form the joint as quickly as possible and then chill by blowing upon it. The joint should be 2¼ to 4 in. long and only slightly thicker than the walls of the pipe.

Care should be taken not to heat the solder above 600 F, as lead melts at 621 F, and the pipe will be injured if the solder is too hot. Other types of wiped joints, such as branches, are merely adaptations of the method just described. For instance, in wiping a branch to a lead pipe, an oval hole smaller than the branch is cut in one wall of the pipe with a tap-borer. The edges of the hole are then flared and the end of the branch beveled to make a tight fit. The mechanics of wiping are the same.

Wiped flanged joints are often advantageously used where lead pipe passes through holes in floors, joists or other structural members and can be made to help support the pipe. A sheet lead collar is fitted around the pipe at the structural member just below the flanged end of lead pipe. Then the two ends of pipe and the collar are wiped together.

Fig. 3—Final wiping of joint. This is the oldest type of lead pipe connection

CHOOSING *the right* AIR CONDITIONING SYSTEM

By A. Warren Canney*

Part 6—Central Station System

WHERE the amount of space to be air conditioned is large—department store, theater, several office floors, an entire structure, etc.—there is a choice between running chilled water to a large number of evaporating units and a centralized conditioning apparatus. If winter conditioning is desired, then the central type is indicated, or if the equipment is to operate during intermediate seasons (but not in winter) the central spray system may show the best investment balance. If only summer conditioning is required, then the local evaporating units may be indicated. Where the space is but a few office floors, let us say, and only summer conditioning is desired, one centrally located evaporating coil might well be considered.

Trend in large office spaces and in theaters is to include winter humidification from central systems, which demands reheating to room temperature. As economic pressure lets up, the inclusion of year 'round conditioning will be more frequent with all types of jobs; while direct radiation is by no means expected to disappear, comfort conditions in buildings will be improved by designing the air conditioning system to supplement direct radiation. For instance, interior spaces are frequently cold due to infiltration from elevator shafts, direct radiation being mostly confined to the perimeter of buildings.

Despite the many applications of unit conditioners and the vast need they meet so well, there is nothing on the scientific horizon which indicates that the central plant with sprays will be displaced for many application conditions in large building areas. It should be borne in mind that with proper design and application, metallic coils for direct expansion or chilled water can and do fulfill building comfort requirements as adequately as sprays. Moreover in actual practice, whether for summer only or the entire year, it is seldom that spray chambers are subdivided in order to yield their prime technical advantage, which is the independent control of relative humidity. Their preference on very large installations is usually because of their efficiency and consequent reduction of space required. Moreover, they are easily cleaned.

But there are times when it is essential to split the number of spray chambers up according to technical requirements; namely, when considerably different influences occur on room dewpoints in different spaces at the same time. One basis for this is the prevention of window condensation in winter while maintaining the desired relative humidity elsewhere in interior spaces, but the injection of outdoor air will do this equally well. The advantage of the dehumidifier split is that this arrangement also permits killing infiltration at the source, summer and winter.

Aside from the matters just discussed, there is no further room for discussion, as a broad thing, between the technical results—differences of central station or unit systems. Units with ductwork may be employed to produce equally satisfactory results as the central station plant for comfort conditioning in so far as the atmospheric results are concerned, assuming the basic differences just explained have been taken into account. To make this difference specific, metallic coils must be employed so as to produce the desired relative humidity when the ratio of latent to total heat emission in the conditioned space is maximum at the time of maximum load, whence any lower ratio value results in a relative humidity below the "pegged" maximum, which is satisfactory to comfort conditioning work. Sprays, on the other hand, more readily meet independent relative humidity control requirements. Dependent on the pumping arrangement, the economy gained by varying the temperature level to suit with sprays and hence refrigeration, reduced operating costs therefrom may be offset by the elimination of a greater differential in the pumping head with closed coils.

Considerations in Choosing a Central Plant

There are five general characteristics of spaces for which central station plants are to be generally considered:

1. If the areas to be served are comparatively large and requiring say, roughly, 50 tons or more.
2. The prospective purchaser either owns the building or has a long lease.
3. Where there is but little likelihood of any extensive future changes in the subdivision of the spaces to be served.
4. Where there is but little likelihood of the heat loads increasing in the individual spaces above that for which the installation will be designed to handle.
5. Where there is little possibility of any sizeable portion of the spaces to be served to be in disuse or unoccupied while conditioning is required for the remaining spaces.

● **This series is addressed to the prospective owners of air conditioning systems, and is intended to point out the factors that must be considered in selecting the proper type of system for the particular application . . . The central station type of system is considered this month.**

*Engineer in charge of air conditioning, Clyde R. Place, Consulting Engineers, New York, N. Y.

Where these conditions obtain, any one or more of the following four factors may decide definitely in favor of the centralized plant:

1 Quieter conditions than either self-contained units or units only with fans would create in the conditioned spaces.
2 Where outdoor air can be secured only from one point.
3 Where maintenance of the units in the conditioned spaces would be deemed objectionable.
4 Where space or architectural considerations favor locating apparatus outside of the conditioned spaces.

Of these nine general rules only Item (4) in the first group can be considered really technical in respect to air conditioning. All the others become technical of course as they affect design. But even though the first group are listed to be usually influencing, the stores in Radio City are served by a special design to meet the requirements of Items (4) and (5) being absolutely unpredictable, much less fixed in advance of design as they should be for central station work. The reason for this was the considerable value of the space as listed under Item (4) in the second group. But this list may well be used as a guide.

Importance of Control

In buying an air conditioning system, particularly of the central plant type, two of the main things to watch out for are securing proper air distribution and proper subdivision of control; and the latter means also a proper subdivision of system layout and design so that it can in fact be controlled.

As a general thing, in buildings which have an appreciable weather load with respect to the total load—as

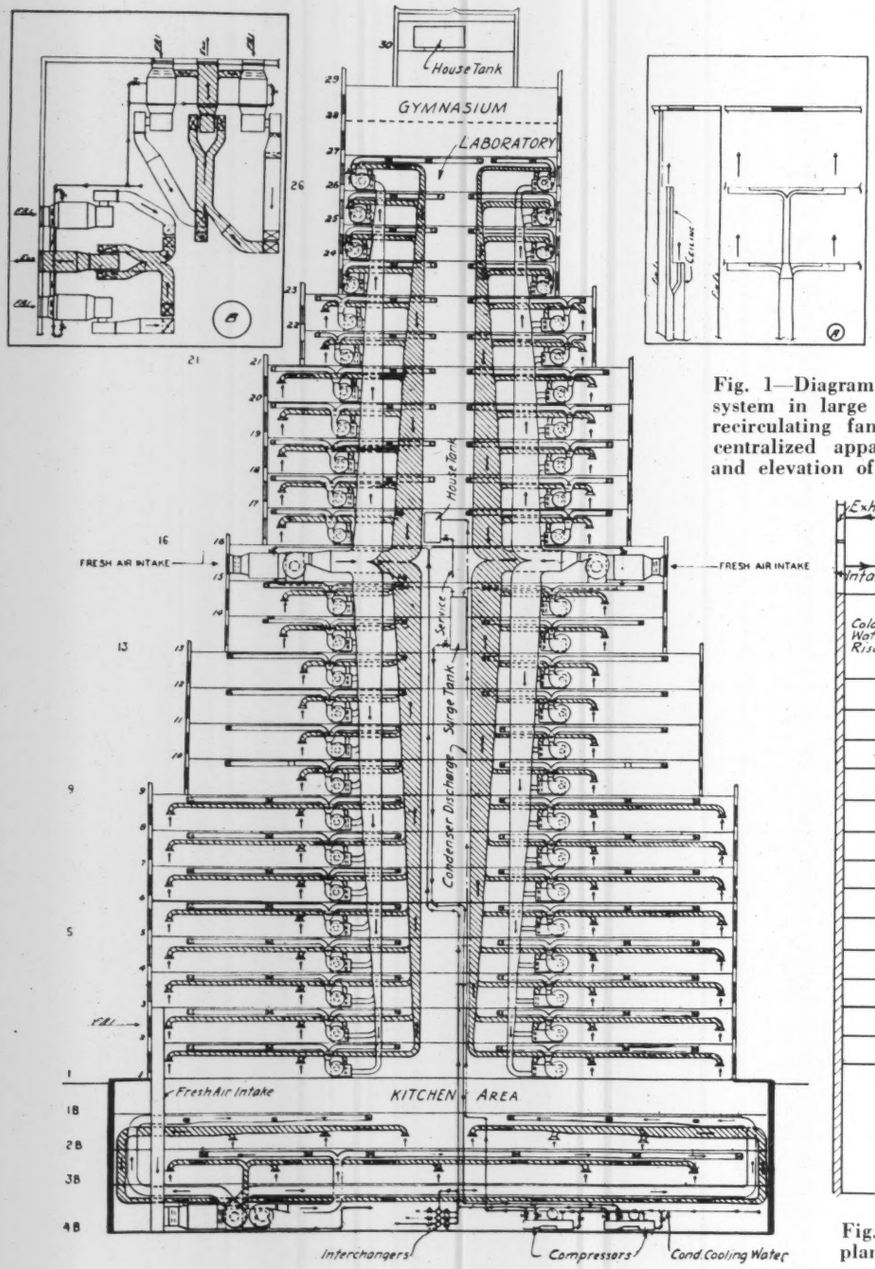


Fig. 1—Diagram of central station air conditioning system in large building. Note that there are two recirculating fans per floor. Sketch B shows the centralized apparatus layout, and Sketch A plan and elevation of typical portion of the duct system

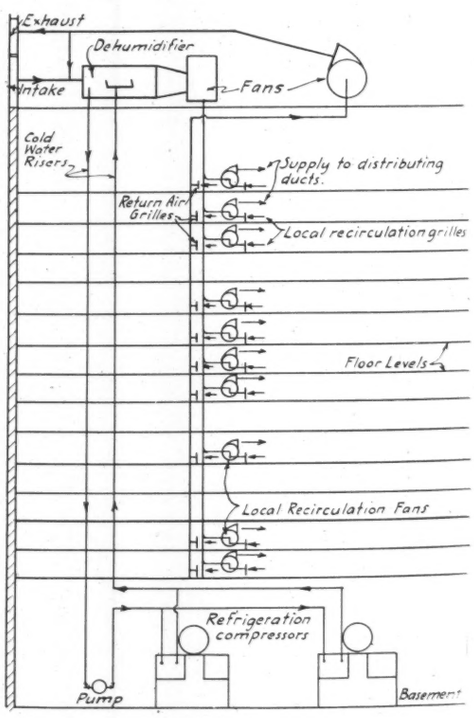


Fig. 2—Diagram of another central station plant, with but one recirculating fan per floor

most buildings with windows do—control is to be subdivided into a minimum of four zones horizontally, south; west; north and east; and interior, this last if there be partitioned off interior spaces. Independent control of individual interior spaces may be essential, dependent on load variation (See June issue, page 288).

Figs. 1 and 2 show two central station plants in diagrammatic form. Both are of the same general design principles but differ in the important respect that Fig. 1 shows two recirculating fans per floor while Fig. 2 but one. In view of what was written above regarding horizontal subdivision for control it should be explained that there are no windows in the west wall of the Fig. 1 building, whereas the building in Fig. 2 has a relatively very low weather load as compared to the total.

Both plants vary the volumes of recirculated and conditioned air by automatic control, keeping the total delivered constant. This is an excellent arrangement and well fulfills all of the requirements. In the use of aspirating outlets the volume of air necessary could be reduced to that of the conditioned air volume, the recirculating fans omitted and replaced by volume dampers. While more economical in first cost, so doing does not always suggest the right air conditioning method, as these plants well typify.

Sketch B shows the centralized apparatus layout for the Fig. 1 plant and sketch A shows a plan and elevation of a typical portion of the duct distributing system.

The importance of control can well be illustrated by the plant in Fig. 1. The dehumidifiers in the basement receive chilled water through interchangers. The building is occupied by over 7,000 people above the street level. The basement systems serve cafeterias having a capacity of over 2,500 people. Thus, at noon, one-third of the population-load shifts from above ground to the cafeterias in about fifteen minutes. When this takes place the problem is to design so as to prevent the cafeteria system from cycling or going into a hunting action with consequent alternating overcooling and overheating.

What the Purchaser Should Look For

This well illustrates that when most prospective purchasers buy air conditioning the time spent in self-education on the manner of determining plant capacity through Btu calculations as a means of checking plant capacity for assurance of minimum price usually detracts from the time one should spend on many other important details best known to those who have been put in the position of making air conditioning systems work. Determination of capacity is a comparatively elementary process. The emphasis should be, rather, on the manipulation of all the technical variables so that all of the requirements of the owner or manager will be met at minimum ownership costs. In choosing a central station plant, local circumstances, the wishes and needs of the owner and architect, have more to do with it than any purely technical considerations pertaining to air or equipment. In this respect, however, it is frequently up to the advisory engineer to know the best method to cover all the best interests of the owner's requirements, the point being, that where general rules are given above, there has been sufficient experience to point out demonstrable instances of exception to all of them. But for the purpose of our series they will serve both as a substantial guide and a starting point.

Progress in A. C. Standardization

ON June 6, twenty business and professional men interested in air conditioning standards met and organized themselves into The Chicago Air Conditioning Advisory Commission. The commission elected John Howatt chairman and authorized him to act as its agent in organizing a committee to be known as the "Chicago Committee on Air Conditioning Standards," the purpose of which committee being to continue the work of the original group that developed the Standards for Certified Air Conditioning for Comfort.

Each of the major groups or associations interested in air conditioning work was invited to appoint one representative to serve on the committee. The first meeting was held June 14, at which time the committee organized, electing the following officers: John Howatt, Chairman; Harry M. Hart, Treasurer; Samuel R. Lewis, Secretary.

The purposes and objectives of the committee were established as follows:

1. Using the Standards for Certified Air Conditioning for Comfort dated February, 1935, as a basis to develop Air Conditioning for Comfort Standards for Chicago that will serve as a guide for air conditioning work in this community.

2. To consider ways and means of promoting and popularizing these Standards.

3. To consider ways and means of making the standards as finally developed effective.

The first objective is being considered first. The committee is meeting regularly and working diligently in an effort to develop standards upon which there can be an agreement and which it will be possible to administer. Insofar as possible it is the aim of the committee to specify definitely only those factors that are measurable. Minimum requirements only will be stipulated and every precaution taken against setting up standards that will stifle ingenuity or progress. Stress will be made on the factors that affect the design load in both heating and cooling.

It is recognized that the primary need for standards is found in the necessity for providing a protection to the customer that does not now exist. Without standards or without a knowledge of their existence the customer may be ignorant of what he is purchasing. He is not aware that air conditioning may be furnished in whole or in part and in different degrees of excellence so is easily misled into considering the lowest bid the best, whereas as a matter of fact it may be the poorest.

The commitee expects to continue its work until a complete set of standards has been agreed upon, when they will be submitted to the different representative groups and associations for consideration.—John Howatt, Chairman, Chicago Committee on Air Conditioning Standards.

How to Reduce NEXT WINTER'S HEATING COSTS

By **J. Earl Seiter***

CORRECT heating conditions are those where there is no sensation of heat or cold by the occupant. That definition covers true comfort from a heating standpoint, or perhaps better stated, proper heating from a comfort standpoint. Further, it signifies a condition where there is peace of mind and no requirement for manual manipulation of radiator valves or windows to provide the degree of comfort desirable. Fortunately, this type of heating service is generally the cheapest to provide and saves money not only in heating costs but in numerous other ways as well. Also, it assures the building of progressive and desirable tenants.

Fifteen years ago heating was simply a method of supplying heat to a building. It was a practical problem and if sufficient heat was always available that was all that could be expected. Since that time heating has developed into a science; the progress made speaks for itself, as savings from 15 to as high as 60 per cent have been made. It formerly was assumed that the steam consumption for heating would amount to an average of 600 lb per sq ft of radiation per yr for average weather conditions in this country; now a figure nearer to 300 lb is accepted.

My experience has been limited principally to buildings heated with district steam service. Many of the principles discussed are applicable to heating systems, regardless of the source of heat used, but many of them are much more effective when district steam service is used. Throughout the discussion it should be remembered that figures represent actual metered results.

Principles of Heat Conservation

In general the principles to be applied in heat conservation involve:

Reducing the amount of heat losses from the building.

Limiting the hours of heating to those in which the required temperature is necessary.

Regulating the amount of heat so as to prevent overheating during the hours of occupancy.

To obtain the full benefit of the application of these principles, it is necessary that the heating system be designed to get good distribution quickly and thoroughly throughout the system and that the heating system and its accessories be maintained in good operating condition.

Glass Losses—A principal source of heat loss is through glass, and is five times as great as through the average wall surface. Therefore, the installation of storm windows on all windows in corridors, stairways, elevator shafts, etc., where it is not necessary to open and close the windows, is recommended. The extra glass reduces the heat loss through the window about 50 per cent.

Roof Losses—Inasmuch as the warmest air in a building is at the top, roof losses are high. It is usually cheaper properly to insulate the roof than to make up the high losses with additional radiation.

Infiltration—There is a certain amount of cold air entering a building around windows and through walls; usually the amount of this air infiltration depends upon the size, number, and type of the windows. Some forms of sash allow an excessive amount of leakage to take place. Such leakage can be very materially reduced by the use of weather strips. They can be applied quite easily and their use shows by actual tests a saving of from 11 to 34 per cent in the heating season's consumption of steam.

Often there is a considerable leakage between the window frame and the building wall. These openings can be closed at a relatively small cost by calking. Skylights, like windows, often leak around the frames. Drafts are caused by the cold air dropping to the floor. This down draft can be remedied by the installation of heating coils around the glass surface, but the installation of double skylights will accomplish the same result and in most cases the cost of the double skylight will be less than the cost of installing the heating coils, especially when the cost of operating the coils is considered.

Another source of infiltration is through the doors entering from the outside. Where there is a large amount of traffic, this becomes very heavy. The installation of revolving doors will reduce this loss materially.

In buildings having freight entrances and other openings where revolving doors cannot be installed, there is constant annoyance due to the continual opening and closing of these large doors, causing drafts to all sections of the floors. Most of this discomfort can be avoided and much heat saved by having vestibules on these entrances so that at no time would there be a direct opening between the inside and outside.

Ventilation—Ventilation is mentioned because it does have a decided effect upon the cost of heating. Ventilation is necessary in many cases, and in these cases its cost is justified, but money has been spent installing ventilating equipment in buildings only to find that its operation was unjustified. Ideas on requirements for ventilation have changed, present standards calling for 10 cfm of outside air per person. Under ordinary conditions the natural infiltration of air around windows is sufficient to supply this amount.

Stack Effect—The unbalanced effect of heating between the upper and lower floors of tall buildings is due to the natural tendency of warm air to rise. Buildings having open stair wells and elevator shafts find this condition very difficult to prevent and it frequently results in excess infiltration and resultant waste. In some buildings horizontal baffling has been accomplished at relatively modest expense, but in general it involves expen-

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● Fortunately, correct heating from the comfort standpoint is economical heating as well. Here are suggestions that will save dollars next winter.

sive changes of construction that are hardly justified by the advantage gained. Under such circumstances the tendency can be greatly retarded if care is taken to see that the pent houses and other roof openings are kept tightly closed and that all windows as far as possible on the upper floors are shut.

Day and Night Temperatures

The temperatures necessary in buildings are quite different for day operation and for night operation. A day schedule must be maintained which will keep the inside at a comfortable temperature whenever there are occupants in the building. On the other hand, the only heating normally necessary at night is to maintain the inside temperature from 40 to 60 F. Because the rate of heat transfer from the building varies directly as the difference in temperature between the inside and outside, the lower the inside temperature the lower the heat loss and heating costs.

Night Heating—The greatest economy in the consumption of steam can be effected by shutting off the heating system at night and on holidays. Steam should be turned off as early in the evening as the use of the building will permit, and should not be turned on again until the following morning except in extremely cold weather. Steam can be turned off in the evening somewhat in advance of the time when a drop in building temperature is permitted, because the radiators will give off heat for a time. In mild weather the building temperature will drop slowly and steam can be turned off earlier than in cold weather.

A study of a particular building with recording thermometers will establish the rate of cooling and heating-up for various outside temperatures and a schedule can easily be prepared for turning steam on and off the building. Here is a typical schedule for an office building providing service from 8:00 a.m. to 11:00 p.m.

Schedule for Hours of Heating

Outside Temperature	Turn Steam On	Shut Steam Off
0 F	Steam on Continuously	
10 F	11:00 p. m.	6:00 p. m.
20 F	1:00 a. m.	5:00 p. m.
30 F	4:00 a. m.	4:00 p. m.
40 F	6:00 a. m.	3:00 p. m.
50 F	7:00 a. m. and 1:00 p. m.	10:00 a. m. and 3:00 p. m.
60 F	7:30 a. m.	9.00 a. m.

Working on this basis the building for which the above chart was made obtained a saving in steam of 14,000,000 lb, or over 26 per cent, the first year it was applied. Where sections of the building require heat at times other than the main building, it saves money to provide a separate line to supply special requirements.

Temperature Control

In general, the duration of the extreme low temperature in any locality does not prevail over 2½ per cent of the total time that heating is required in a year. It is

General Rules for Economical Heating

1. Weatherstrip all windows, and calk all window frames.
2. Provide revolving or vestibule doors on all entrances. Separate shipping and receiving rooms by partitions so that the ever-open large doors will not ventilate the entire building.
3. Keep the radiation near the outside walls, under the windows, if possible.
4. Eliminate all unnecessary ventilation. Ventilating equipment is sized to meet extreme requirements. Don't supply ventilation to a theater or auditorium adequate for an audience of two thousand when there are only two hundred present.
5. Determine the hours that heating is required during the day and see that the steam is shut off for the maximum time at night, on Sundays, and holidays.
6. Shut steam off entirely in unoccupied sections of the building, taking care to avoid freezing of plumbing.
7. Shut off steam during the day whenever possible. During the year steam can be shut off about 55 per cent of the total daytime, the saving is proportional. An automatic control will do it perfectly, but it can be done by hand with good results.
8. Determine the temperature required for the occupancy of the building. Do not heat a storage garage or a furniture warehouse to the temperature required in a hospital ward.
9. Provide some good means of temperature control. No building can afford to be without such a control.
10. In a hot water heating system keep the temperature of the water down to correspond with existing outdoor temperatures.
11. In a vacuum system maintain a high vacuum. If this is not possible, locate and eliminate all leaks.
12. Install separate lines for those parts of the building that require long-hour or all-night heating. It is much cheaper than heating the entire building all night.
13. See that the entire system responds rapidly when steam is turned on. Locate and eliminate the cause of any sluggish circulation. Balance the radiation, provide adequate air elimination, and correct any trapped run-outs to provide quick system drainage.
14. Keep the system in good repair. Worn, damaged, or defective valves and traps will not function properly.
15. Cover all steam pipes.
16. Do not cover or otherwise obstruct the free circulation of air around the radiators; to do so seriously reduces the heating capacity of a radiator.
17. Use the heat in the condensate for hot water or some other useful purpose.
18. Provide thermometers and recording pressure gages so that the system can be operated with full knowledge.
19. Make all valves and controls convenient and accessible, either directly or through remote control. It is only human nature to delay and avoid doing that which is unnecessarily inconvenient.
20. Keep a consistent daily record, based on weather requirements, and watch it every day.
21. Know the system and understand its functions and its operations.
22. Control the heat supplied to water tanks located on or above the roof. Such tanks require heat to prevent freezing. No heat is required when the outdoor temperature is above 32 F.
23. Investigate every complaint of "No Heat," find the cause and correct it. Do not overheat an entire building to correct a local condition in one room.

obvious that during the remaining 97½ per cent of the time that the release of heat must be retarded, or in some way controlled, to prevent overheating and the resultant waste. It is fully possible with an average outside temperature of 47 F to waste 67 per cent of the heat used in a building, and it is reasonable to believe that with such wastes possible every effort should be made to provide necessary facilities to eliminate it, and see to it that these facilities are continuously and effectively used to the best advantage.

Excess window openings are not usually the result of a need for ventilation but are a sign of the need for relief from overheating within the building. Time and again it has been proved that the seeming necessity for open windows and consequent fuel waste quickly vanishes when the heating system of the building is brought under control and the temperatures are maintained within the comfort range. Further, outside air in congested business or industrial areas is contaminated by dust, fumes, gases, and smoke which is harmful to buildings and the health of the occupants. It is this dirt coming in through open windows which causes much cleaning and maintenance expense.

One more benefit of temperature control is the elimination of fluctuating building temperatures, caused by manipulation of radiator valves, opening and closing of windows, uneven firing, or many other causes. Health authorities suggest that fluctuating temperatures favor bacterial activity and play an important part in the inception of the common cold and other respiratory diseases. Working in an overheated room results in mental fatigue, whereas with suitable temperature conditions the mind is alert and capable of greater accomplishment.

Modern temperature control assures many benefits which, briefly summarized, are:

For the Building Owner

It saves money by reducing heating expense.

It saves money by reducing cleaning and maintenance expense.

It promotes the peace of mind and the good will of tenants.

It raises the prestige of the building.

It will increase the revenue from the building.

For the Tenant

It leads to more efficient work.

It reduces the amount of sickness.

It provides greater comfort and peace of mind.

It makes possible cleaner and more pleasant surroundings.

What Can Be Accomplished

What can be accomplished by a careful survey of conditions and the application of the principles for conservation of heat is very apparent in the results obtained in a building having approximately 5,500,000 cu ft of space and 68,000 sq ft of radiation. The first floor is ventilated, and on the upper floors vitiated air is removed by an exhaust system. Portions of the building are used 24 hr per day and on Sundays. This building used approximately 35,000,000 lb of steam per season. Radiators were turned off at night by the charwomen and on again by a watchman. About 60 per cent of the space was occupied from 8:00 a.m. to 11:00 p.m. and the balance from 8:00 a.m. to 6:00 p.m.

During the first year of the study, separate lines were installed to the various sections of the building requiring heat 24 hr per day. The portion of the heating system in the building operated from 8:00 a.m. to 6:00 p.m. was segregated from the balance of the system. Steam was shut off completely on the various sections when they were not occupied. The steam consumption dropped from 35,000,000 lb to a little more than 22,000,000 lb.

RECORD OF HEATING

Building ..... Degree Day base 65 F .....

Month ..... Meter Constant .....

Date	Mean Temp. From Weather Bureau	Degree Days 65 Minus (2)	Meter Reading From Meter	Steam Consumption Meter Constant Times Difference	Lb of Steam per Deg Day (5) Divided by (3)	Weather Conditions
(1)	(2)	(3)	(4)	(5)	(6)	(7)
1						
2						
3						
4						
5						
27						
28						
29						
30						
31						
Month						
Lowest Previous Record						
Previous Month						
Previous Year						

Heating Efficiency = Lowest Previous Record (Item 6) / Record This Month (Item 6)

During the second year further corrections were made to improve the circulation in the heating system and the condensate from the building was used in the ventilating equipment. These small adjustments resulted in a further reduction of nearly 2,000,000 lb. The third year temperature control was installed, and the steam consumption dropped to 16,000,000 lb. The fourth year with better experience in operation, and a few minor changes in the system, a further reduction to 14,500,000 lb was obtained. These changes resulted in a total drop in steam consumption from 35,000,000 lb per year to 14,500,000 lb, a reduction of over 60 per cent. Heating conditions were greatly improved and the health record of the building, of which an accurate record was kept, immediately showed a decided improvement. The cost of the changes amounted to approximately $18,000, which was saved in a few years.

It is not always necessary to spend a great deal of money on equipment to make savings, especially if the heating system is in good condition. Shutting off at night requires no capital outlay and some degree of temperature control can be accomplished by manual operation of the steam supply valves. The cost of caulking around windows usually pays for itself in reduced cleaning and painting costs. If a vacuum system is installed in the building, the temperature of the steam in the radiators and heat emission from them may be substantially reduced by maintaining vacuums as high as possible at the reducing valve and in the radiation. Manual operation requires strict supervision of building temperature. This always pays.

Types of Temperature Control

There are many systems of temperature control; the three general types are—

Those providing intermittent operation.

Those providing continuous heating with radiators partially filled with steam, known as the orifice system.

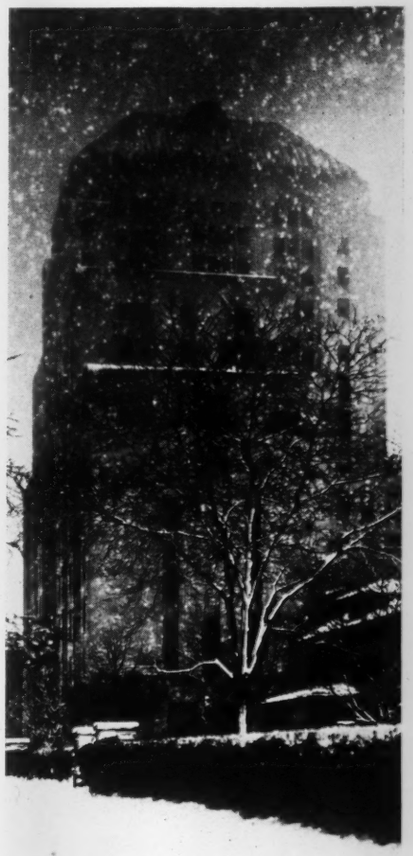
The high vacuum systems.

There is advantage in having the control operated by outside temperatures because of the difficulty of obtaining representative thermostat locations in the building. For small buildings, or for small zones in larger buildings, intermittent type controls are available with this feature. They can be used on either one pipe or two pipe systems. Usually they are relatively inexpensive to install, but require rapid circulation in the heating system and good venting. On larger buildings the use of the orifice system has been extremely successful. It operates from an outside thermostat, requires no vacuum pump and is simple to operate. It can be used on both one pipe and two pipe systems. The high vacuum type of control is primarily applicable to large vacuum systems. It requires the operation of oversized vacuum pumps, but has certain advantages in that the radiator is usually completely filled with steam at temperatures required. It does not have outside thermostats and requires some supervision.

With any type of control equipment a careful engineering survey of the system should be made.

Up-to-Date Knowledge of Building Conditions

The engineer is naturally interested in the large proportion of the building operating cost that is annually charged to the heating system. It deserves, therefore, and should receive a generous part of the engineer's attention and activity in the operation of the system. The tenants of a building, whether it be commercial or residential in character, insist on receiving the proper heating service to which they are entitled and for which they pay. The engineer must keep this in mind, and at the same time must discharge his responsibility to the owner



Money will be saved next winter if attention is given now to plant and building heating systems and control

of the building. He is obliged, therefore, to be constantly aware of all physical conditions pertinent to the operations and to the mechanical equipment with which he operates. He should be so equipped as to have a constant check on both inside and outside temperatures. Recording thermometers are inexpensive and serve an excellent purpose. He should as well be acquainted with the per cent of occupancy of the building, whether or not the radiation in unoccupied space is for the most part shut off and what the varying requirements of tenants or sections of the building are, so that he may serve the system efficiently and at an economical rate.

In the operation of a heating system, actual records of the cost and the factors making up the cost are essential to economical operation. Continual checking and recording of operating data, supplemented by constant endeavor to improve conditions, cannot help but reflect in lowered costs.

Often, however, the heating system, from the stand-

(*Continued on page* 388)

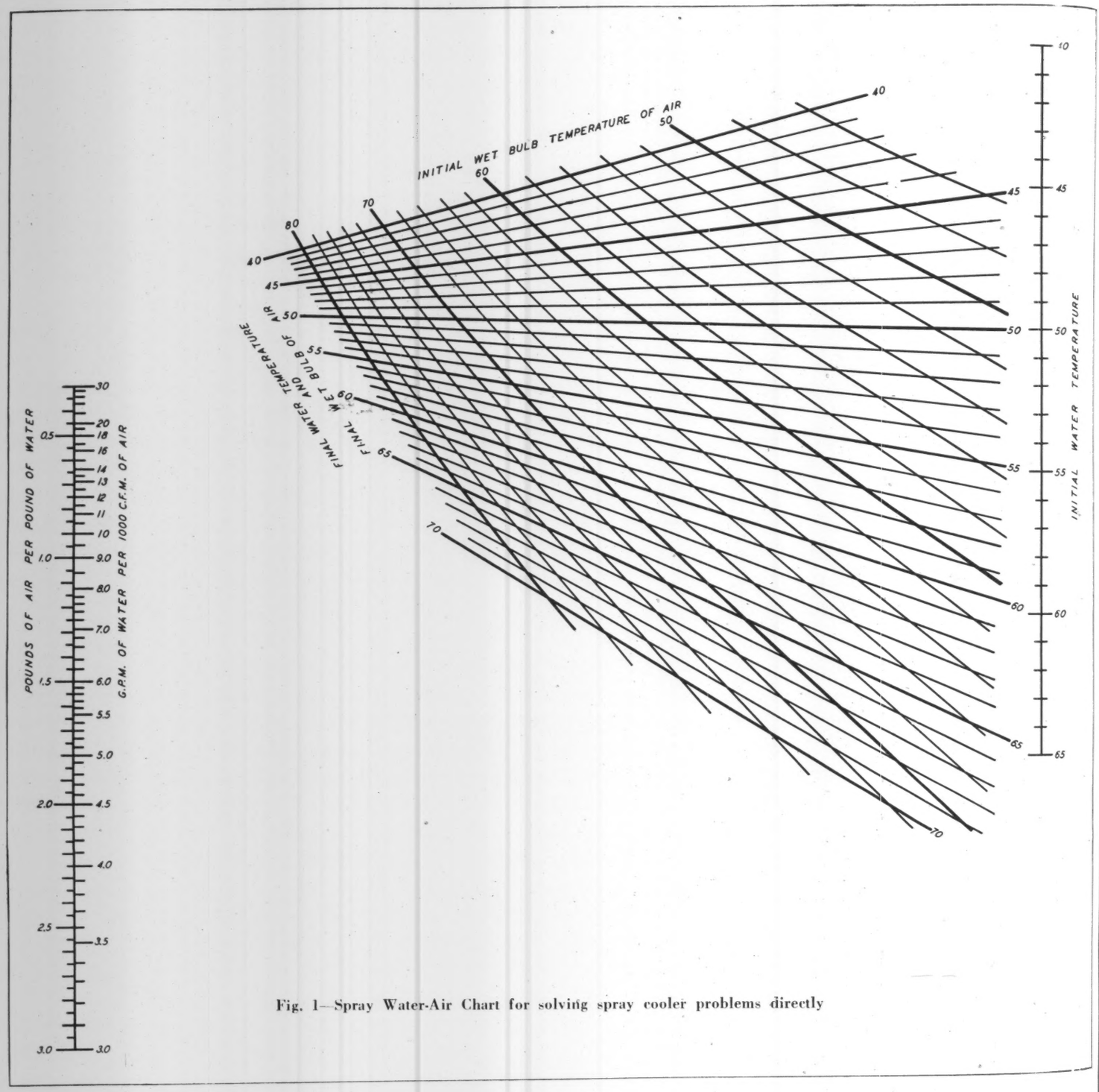


Fig. 1—Spray Water-Air Chart for solving spray cooler problems directly

Two Stage Air Washer Problems Solved by Chart†

By **William Goodman***

• Describing the use of a chart developed by the author to facilitate solution of process and comfort air conditioning problems involving two stage spray coolers or air washers . . . . Basis of the chart and its use with single stage spray cooler problems was discussed in a previous issue.

NUMEROUS problems in connection with spray water type coolers in process and comfort air conditioning can be solved only by trial and error methods. The author therefore prepared the chart, Fig. 1, which enables direct solution of many such problems. Basis of this chart was explained in the May issue,[1] and solutions of problems involving single stage air washers were shown. The present discussion concerns problems involving two stage air washers.

Two Stage Counterflow Coolers

In solving problems involving spray water air coolers, there are five factors involved:

1. Initial air temperature.
2. Final air temperature.
3. Initial water temperature.
4. Final water temperature.
5. Weight of water per unit weight of air.

In the special case of the single stage washer[1], these factors reduce to four, as the final air and water temperatures are theoretically the same. In the two stage counterflow washer, the final temperatures of the air and water are not the same.

Two stage counterflow air washer problems can be reduced to four general classes:

Class	Given	To Find
(1)	Initial air wet bulb temperature. Final air wet bulb temperature. Lb of water per lb of air.	Initial water temperature. Final water temperature.
(2)	Initial water temperature. Final air wet bulb temperature. Lb of water per lb of air.	Initial air wet bulb temperature. Final water temperature.
(3)	Initial air wet bulb temperature. Final air wet bulb temperature. Initial water temperature.	Lb of water per lb of air. Final water temperature.
(4)	Initial air wet bulb temperature. Initial water temperature. Lb of water per lb of air.	Final air wet bulb temperature. Final water temperature.

Note that the final water temperature is always one of the unknowns to be found, and that the four classes of problems are distinguished according to which one of the remaining four factors is unknown and must be found.

Class 1 and 2 problems are easily solved directly by means of the Spray Water-Air Chart. Class 3 and 4 problems can be solved only by trial and error, but this becomes comparatively simple with the chart. The solution of a few problems will make one adept in the handling of problems in the latter two classes.

Fig. 2 is a diagram of a two stage counterflow air washer. Here, as in the case of single stage washers, the guiding principle in the solution of problems is: The temperature of the water leaving either of the washer stages is equal to the final temperature of the air leaving that stage. In solving a problem involving a two stage washer, consider the two stage washer as two separate single stage washers with the air flowing first through one and then through the other. Referring to Fig. 2, note that:

(a) The final temperature, t'_2, of the water leaving the second stage is equal to the final temperature of the air leaving the cooler.

(b) The final temperature, t_2, of the water leaving the first stage is equal to the temperature of the air leaving the first stage.

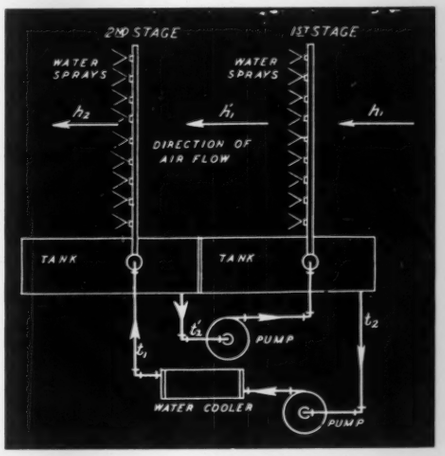


Fig. 2—Two stage counterflow spray cooler

Problem 1: (Illustrative of Class 1 Problems)

See Fig. 3.

Initial wet bulb of air is 68 F.

Air to be cooled to 50 F.

Weight of water supplied is 8 gpm per 1000 cfm of air.

Find the required initial temperature of the water and its final temperature.

If the air is cooled to 50 F, then evidently the temperature of the water leaving the second stage and entering the first stage is 50 F. Using 50 F as the initial temperature of the water entering the first stage, find the temperature of the air leaving the first stage by treating the problem as if the first stage were a single stage washer alone. The temperature of the air leaving the first stage will be 58 F, which is also the final temperature of the water leaving the cooler.

Now, using 58 F as the initial temperature of the air and 50 F as the final temperature of the air, solve for

* The Trane Co., La Crosse, Wis.
† Copyright, 1935, by William Goodman.
[1] See H. P. & A. C., May, 1935, pp. 232-235.

the initial water temperature exactly as if the second stage of the washer were a single stage washer alone (See H. P. & A. C., May, 1935, pp. 232-235).

The inlet temperature of the water must be 44.8 F.

Check:

Heat gained by water = Heat lost by air.

8 gpm = 0.89 lb of water per lb of air.

Heat gained by water = 0.89 (58-44.8)

= 11.74 Btu per lb of air cooled.

Heat lost by air in cooling from 68 F wet bulb to 50 F wet bulb = 31.92 — 20.19.

= 11.73 Btu per lb of air cooled.

Problem 2: (Illustrative of Class 2 Problems)

See Fig. 4. Water having an initial temperature of 45 F is to be used to cool air. If 12 gpm of water per 1000 cfm of air is supplied, what is the highest initial wet bulb temperature which the entering air may have if this air must be cooled to a final temperature of 48 F saturated?

Evidently, since the final air temperature, the initial water temperature, and the weight of water are given, the wet bulb temperature of the air entering the second stage of the cooler can be found by means of the Spray Water-Air Chart.

From the chart, the wet bulb temperature of the air entering the second stage is 55 F, which is also the final temperature of the water. If the air is cooled to a final temperature of 48 F, then the temperature of the water leaving the second stage and entering the first stage must be 48 F.

Using 48 F as the initial water temperature and 55 F as the final air temperature leaving the first stage, it is found from the chart that the highest wet bulb temperature which the air entering the first stage may have is 69 F.

Check:

$$\frac{12 \text{ gpm}}{9} = 1.33 \text{ lb of water per lb of air.}$$

Heat gained by water = 1.33 (55—45)

= 13.33 Btu per lb of air cooled

Heat lost by air in cooling from 69 F wet bulb to 48 F wet bulb = (32.71—19.12)

= 13.59 Btu per lb of air cooled.

Problem 3: (Illustrative of Class 3 Problems)

Air having an initial wet bulb temperature of 74 F is to be cooled to a final temperature of 46 F. The initial water temperature is 42 F. Find the gpm of water required and its final temperature.

The simplest way of solving a problem of this class is to assume a given gpm per 1000 cfm of air and solve this problem as if it were a Class 1 problem.

A few trial attempts with various water quantities will yield an initial water temperature equal to the initial water temperature given in the problem.

Attempt (a)

Assume 10 gpm per 1000 cfm of air.

Then t_2 = 57.5 F final water temperature and t_1 = 40.3 F required initial water temperature.

Therefore 10 gpm of water is not sufficient since in order to utilize this quantity its initial temperature must be 40.3 F or less while the conditions of the problem state that water no lower than 42 F is available.

Attempt (b)

Assume 12 gpm per 1000 cfm of air.

Then t_2 = 56 F final water temperature and t_1 = 41.8 F required initial water temperature which agrees closely with the initial water temperature of 42 F given by the problem.

Therefore 12 gpm of water per 1,000 cfm of air is required and the final temperature of the water will be 56 F.

Check:

$$\frac{12 \text{ gpm}}{9} = 1.33 \text{ lb of water per lb of air.}$$

Heat gained by water = 1.33 (56—41.8) = 18.92 Btu per lb of air.

Heat lost by air = (36.91—18.09) = 18.82 Btu per lb of air.

Problem 4: (Illustrative of Class 4 Problems)

Fifteen gpm of water having an initial temperature of 50 F is available. Air having an initial wet bulb

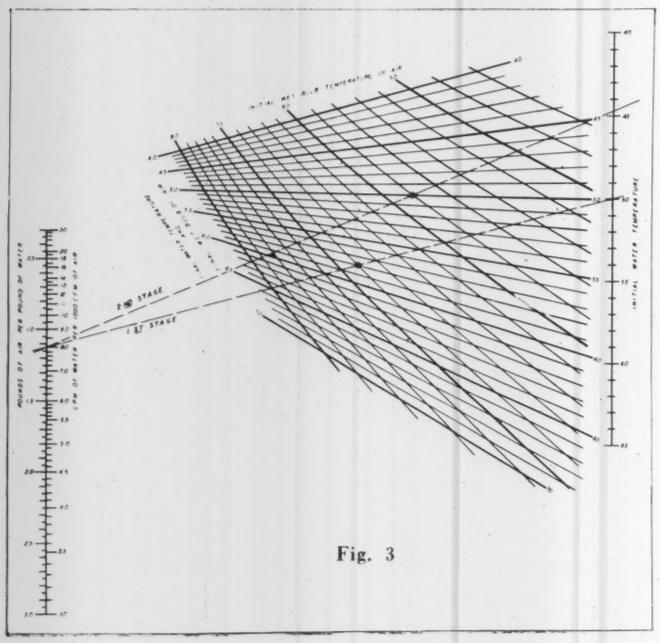


Fig. 3

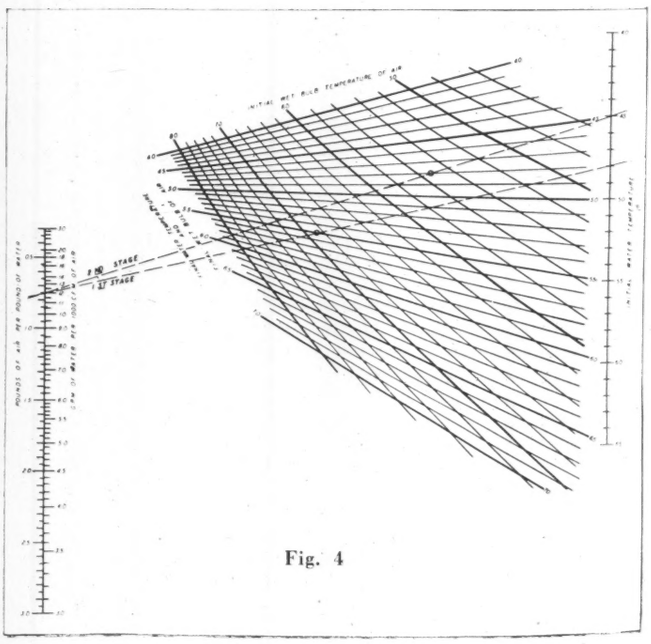


Fig. 4

temperature of 69 F is to be cooled by this water. What will be its final temperature?

The simplest way to solve problems of this class is to assume the final air temperature and solve for the initial water temperature as if this were a Class 1 problem, as in the previous example. A few attempts with various final air temperatures will yield an initial water temperature equal to the one given in the problem.

Attempt (a)

It is evident that the final air temperature must be above the initial water temperature and because of the large amount of water available, it will be only a few degrees higher than the initial water temperature.

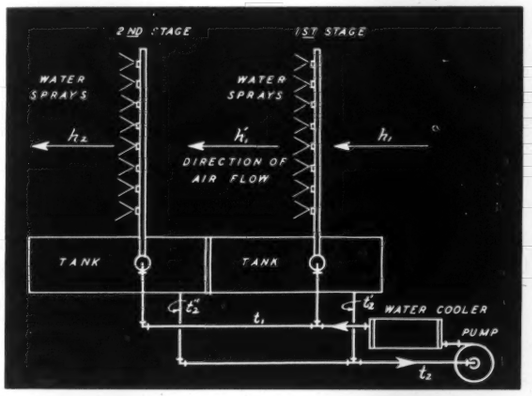


Fig. 5—Two stage split spray cooler

Assume 53 F as the final temperature of the air.

Then $t_2 = 57.8$ F, the final water temperature and $t_1 = 51.4$ F required initial water temperature.

Too high a final air temperature has been assumed since the initial water temperature required is higher than 50 F. That is, to cool the air to 53 F requires water having an initial temperature of only 51.4 F while 50 F water is available. This means that the air can be cooled to less than 53 F.

Attempt (b)

Assume 52 F as the final temperature of the air.

Then $t_2 = 57.1$ F, the final water temperature and $t_1 = 50.2$ F, the required initial water temperature which agrees closely with the initial water temperature of 50 F given by the problem.

Therefore, the final air temperature will be 52 F and the final water temperature 57.1 F.

Check:

$$\frac{15 \text{ gpm}}{9} = 1.67 \text{ lb of water per lb of air.}$$

Heat gained by water = 1.67 (57.1—50.2) = 11.5 Btu per lb of air.

Heat lost by air = (32.71—21.30) = 11.41 Btu per lb of air.

The pump shown in Fig. 2 which delivers the water from the second stage tank to the nozzles of the first stage has some heating effect on the water. However, with the quantities of water and the heads usually involved in air conditioning, the temperature rise of the circulating water, due to the heating effect of the pump, will as a rule be less than 1 F.

Two Stage Split Air Washers

In the two stage split washer, the water is divided or split between two banks of sprays. Fig. 5 is a diagram of such a cooler; as shown, the water going to both stages is at the same temperature. It makes little difference in what relative proportions the water is supplied to the two stages if one stage gets at least 25 per cent of the total. The air is cooled to the lowest point when the water is equally divided between the two stages. However, as can be seen in Table 1, when the water is divided between the two stages in a 25-75 per cent ratio, the final air temperature is less than ½ F higher than if the water were equally divided between the two stages.

Problems involving the two stage split cooler are solved by treating the cooler as if it consisted of two simple single stage coolers with the air flowing first through one and then through the other. The wet bulb temperature of the air leaving the first stage is found first, and then using this as the initial wet bulb temperature of the air entering the second stage, the final temperature of the air can be found. The following example illustrates the solution of problems involving these washers.

Problem 5:

Sixteen gpm of water having an initial temperature of 50 F is supplied to a two stage split cooler. To what final temperature can air having an initial wet bulb temperature of 74 F be cooled? If the water is split equally between the two stages, 8 gpm is supplied to each stage.

Answer:

The air leaves the first stage at a temperature of 61.1 F. This, therefore, is the initial wet bulb temperature of the air entering the second stage. The final temperature of the air leaving the second stage will be 54.5 F.

Check:

$$\frac{8 \text{ gpm}}{9} = 0.889 \text{ lb of water per lb of air.}$$

FIRST STAGE

Heat gained by water = 0.889 (61.1—50) = 9.85 Btu per lb of air.

Heat lost by air = (36.91—26.91) = 10.0 Btu per lb of air.

SECOND STAGE

Heat gained by water = 0.889 (54.5—50) = 4.0 Btu per lb of air.

Heat lost by air = (26.91—22.75) = 4.16 Btu per lb of air.

Table 1—Effect on the Final Temperature of the Air Leaving the 2nd Stage of a Two Stage Split Air Washer When Varied Proportions of the Constant Total Water Supply are Sent to Each Stage Instead of Being Split Equally Between the Two Stages (Based on a Total Water Supply of 30 gpm per 1000 cfm of Air)

GPM of water to first stage	Per cent of total water supply sent to 1st stage	Final Temperature of the Air When Initial air wet bulb = 74°F. Initial water temp. = 50°F.	Initial air wet bulb = 74°F. Initial water temp. = 42°F.	Initial air wet bulb = 64°F. Initial water temp. = 50°F.
4.5	15	52.6	45.2	51.3
6.0	20	52.4	44.9	51.2
8.0	26.67	52.2	44.6	51.1
10	33.33	52.1	44.4	51.0
12	40.0	52.0	44.3	51.0
14	46.67	51.9	44.2	50.9
16	53.3	51.9	44.2	50.9
18	60.0	52.0	44.3	51.0
20	66.67	52.1	44.4	51.0
22	73.3	52.2	44.6	51.1
24	80.0	52.4	44.9	51.2
25.5	85.0	52.6	45.2	51.3
30.0	100.0	54.2	47.5	52.2

SELECTION CHART FOR COOLING COILS

Psychrometric Chart
Wet Bulb and Temp. Coil Surface (t_2)
Dry Bulb Temp. deg. F
(To determine Temp. Coil Surface Required)
Total Heat - Btu per lb dry air
Reduction in dry bulb Temp desired ($t_1 - t_2$)
(To determine no. tube rows required)
DIRECT EXPANSION COOLING COILS
(Velocity thru. face area = 400 f.p.m. 70° F air)
2 ROWS
3
4
5
6
Total Heat Rating of Cooler Btu. per 100 c.f.m.
(Multiply by 4 for Rating per Sq. Ft. of Face Area)
Temperature Coil Surface (t_2)
Temperature of Air (t_3)
Difference
5/8" Copper Coils - 8 fins per inch.

Air Conditioning with Ice

Consumption of Ice and Load Factor for Several Installations

By **Clifford F. Holske***

IN 1931, the American Ice Company became actively interested in air conditioning because of the apparent possibilities of this field as a market for ice. Because of the unsatisfactory business condition during the depression years this market has not developed as rapidly as we had expected. However, we have never lost our faith or interest in air conditioning and we have been able to accumulate some very interesting figures covering the ice consumption in many of the air conditioning installations in our territories during the past four years.

Early in 1932 an "iced air" system was installed in one of the Childs restaurants in New York City. This restaurant has a seating capacity of 400, the calculated refrigeration load being 35 tons. Like most ice systems this one was designed for operation with ice about 50 days per year, and sufficient fan capacity was installed to secure a relatively rapid air change in order that satisfactory conditions might be maintained during the balance of the summer without the use of ice. During the summer of 1932, the ice consumption was 300 tons, in 1933, only 255 tons were used and in 1934 the consumption increased to 347 tons.

The first check that we can make is the number of ice deliveries per season in order to find out how close this 50 day figure, which is so commonly used in ice cooled air conditioning calculations, approaches the actual condition attained in the field. This figure is of importance as it is the basis for most estimates of ice consumption. For instance, an estimate of the ice consumption on this job would probably have been prepared on this basis: 35 ton load x 50 days ÷ 2 (12 hr operation) = 875 tons. If we assume that due to the nature of the business the rooms would be filled to capacity for not more than two hours per day and that for the rest of the operating day the loading would be 25 per cent of the calculated peak we arrive at a net average loading for the period of operation of 37½ per cent. 875 tons x 37½ per cent = 328 tons of ice. Without the 50 day approximation of operating time we would find it difficult to arrive at a rational estimate of ice consumption.

Our records show that in 1932 ice was delivered to this job on 58 days, and in 1933 on 54 days. As many of the deliveries were considerably below the calculated amount required it would seem that the 50 day figure may be considered as being quite conservative. Unfortunately, the number of deliveries in 1934 was considerably above the number made in either of the other two years, but as the total ice consumption for that year does not greatly exceed the estimated figure, I think that we may attribute the increase to the thrift of the management, particularly as the cost of ice for air conditioning is considerably below the cost of ice sold for other uses.

The ice meltage data will also give a very fair idea as to the load factor of the refrigerating equipment used in air conditioning. The term "load factor" in this connection is taken to mean the ratio between the refrigeration actually consumed per year and the refrigeration that would be consumed in the same period if the peak load were applied 24 hours a day, 365 days per year. The conception of load factor is of great importance in evaluating the worth of the various available forms of refrigeration for any particular application.

Economically the status of ice cooled air conditioning is largely dependent upon the load factor. In these systems the greatest single item of cost is the ice bill, and where conditions make it necessary to operate the refrigerating equipment of an air conditioning system during a large part of the year, the ice system will not be economical. Where low load factors prevail the interest and depreciation charges on mechanical refrigerating equipment when prorated over the few hours of operation will frequently exceed the cost entailed in melting ice to produce an equivalent amount of refrigeration.

For this installation, we note that the load factor for the year 1932 was evidently 2.4 per cent; in 1933, 2.05 per cent and in 1934, 2.8 per cent.

There is one other conclusion that can be reached

*Engrg. Dept., American Ice Co., New York, N. Y.

Selection Chart for Cooling Coils

By **Louis A. Harding***

The procedure for use of the direct expansion coil chart shown on the opposite page is as follows:—

1st. Locate initial and final condition of air entering and leaving cooler on psychrometric chart *a*. Draw a straight line between these two points (cooling path) extended to the intersection with the saturation line as indicated. This intersection determines the temperature of coil surface *tx* required.

2nd. Determine the number of tube rows depth of cooler required from chart *b*.

3rd. Divide the total heat Btu per hr to be removed in cooler by the cu ft of standard air (70 F) divided by 100. Refer to chart *c* and locate the intersection of this quotient with the number of tube rows previously determined by chart *b*. The intersection, chart *d*, of the vertical line through the required value of *tx* with the horizontal line drawn through the intersection from chart *c* gives the required temperature of the refrigerant *t* to be employed.

The data from which this chart was plotted was published in the writer's article in the July, 1935, H. P. & A. C., pp. 317-322. It may be used for the selection of direct expansion finned copper tube cooling coils for any operating condition.

*L. A. Harding Construction Corp., Buffalo, N. Y.

from the total tonnage figures of this installation. You will note that there is a considerable variation from year to year, the 300 tons figure of 1932 dropping to 255 tons the following year and then increasing to 347 tons in 1934. There may be other contributing causes, but I believe that the primary reason for this variation could be found in the records of the volume of business handled in that restaurant. If those records were available, I believe we would find that the ice consumption varied quite regularly with the volume of business transacted during the three cooling seasons, and this is another item of merit peculiar to the ice cooled system. When the need for refrigeration drops off the cost of refrigeration drops with it.

Another record of interest is that of the air conditioning system in the Occidental Hotel in Washington, D. C. Dining rooms having a seating capacity of 250 people are conditioned. This installation was figured for eight hours of operation per day and consumed 375 tons of ice in 1932, 450 tons of ice in 1933 and 416 tons in 1934. The number of ice deliveries per season was 67 in 1932, 58 in 1933 and 62 in 1934.

An instance of extremely low load factor is shown in the record of the 125 seat Huyler's restaurant in New York City in which the cooling system was designed to operate not more than four hours per day. The installation has a nominal capacity of 15 tons and in 1932 used only 110 tons of ice for the entire year. In 1933 this dropped off to 70 tons and in 1934 to 67 tons.

The "iced air" system at the 600 seat Stanley theater in the heart of the Times Square section of New York City was also completed in 1932. During the first season 265 tons of ice were consumed, in 1933 the consumption increased to 340 tons and in 1934 dropped to 217 tons. The calculated refrigeration requirement of the Stanley was 50 tons. In 1932, 34 deliveries were made and in 1933 the same number, but in 1934 the figure dropped to 24. There is a considerable variation from year to year in all of these ice consumption figures but not a greater variation than we can reasonably attribute to the variations in loading which occur in every business.—From a paper presented at a meeting of the Air Conditioning Bureau, Boston, Mass.

Heating Costs—

(*Continued from page* 381)

point of economy, receives the least attention in its operation. A simple form or log can quickly be devised which, if properly kept, will furnish the engineer and the building manager with an intelligent indication of the operation and costs.

Since there is a direct relation between the steam consumption and the difference in temperature between the inside and outside, it is of importance that a record of these temperatures be maintained.

Heating Efficiency—On a cold day when temperature is at or below that for which the heating system was designed, there should be very little heat wasted in the building, because the building cannot be overheated to any great extent. Now let's do a little figuring—take the steam consumption for that day and divide it by the difference between 65 and the mean outside temperature. This will give the steam consumption per degree per day under the most favorable conditions for the greatest possible economy in operation. Such units of usage may be considered as 100 per cent heating efficiency.

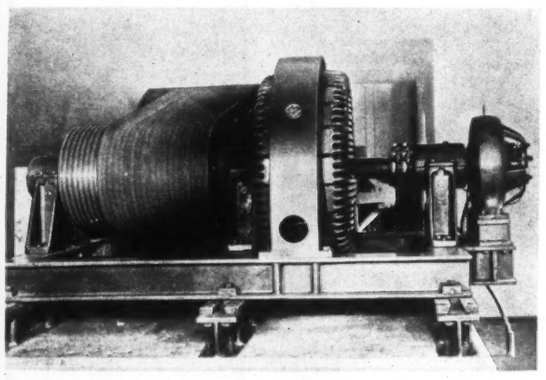
It will be found that the units of usage for other days will be slightly higher, or in other words, the heating efficiency will be lower. The ratio of the lowest unit usage in the heating season to the daily unit usage will represent the heating efficiency for the various days.

This heating efficiency is comparable with those of other buildings, and offers fair comparisons on several buildings, and at the same time overcomes the objections to comparisons that are based on rentable area, radiation, volume, and so forth.

Other factors, such as wind, humidity, solar radiation, shadows, and the like, have distinct bearing on the heating requirements for any specific day's study. While no general method of computing these effects has been accepted, usually the building engineer through experiment and experience will quickly determine the effects and be guided accordingly.

100% Return on Modernization Investment

The Pond Creek Pocahontas Company required additional air for its ventilating system and a means of improving its load power factor. Selected to replace the old drive, a 600 hp synchronous motor with related



Synchronous motor which paid for itself in one year by improving power factor

control now supplies ample ventilation at all levels of the 650 ft shaft, and the needed corrective kv-a to the power system. This change, resulting in a saving of $1,000 a month, paid for the new equipment in the first year of operation, according to the coal company.—R. W. Orth, General Electric Co.

Heating · Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers JOURNAL SECTION

AUGUST 1935

Table of Contents

A Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads, by F. H. Faust, L. Levine and F. O. Urban....391

An All Electric Heating, Cooling and Air Conditioning System, by Philip Sporn and D. W. McLenegan....402

Proceedings of the Semi-Annual Meeting 1935....410

Candidates for Membership....418

Candidates Elected....418

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A Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads

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Introduction

SCIENTIFIC application of air conditioning equipment requires that such equipment shall have adequate capacity to maintain specified indoor conditions, but that the margin of capacity over what is required shall not be so great as to make the installation economically unjustifiable. Thus, the first step in an application is to determine the maximum cooling effect which is necessary, and the second step is to select equipment which is capable of producing this required cooling effect both economically and with a degree of flexibility necessary to meet varying conditions of operation.

This paper summarizes the factors which affect the cooling load, and describes a systematic and rational method for accurately determining its character and magnitude. A heat gain calculation sheet is presented which suitably organizes and tabulates the work in a minimum of space.

The heat gain method to be described presents certain features of advantage to the user, namely;

1. It is applicable to any kind of an enclosure, ranging from a single room to a complete building.
2. It saves time required for making the calculations.
3. It minimizes the chances of error by systematizing the work and providing a suitable calculation form.
4. It permits the engineer to select equipment of adequate capacity, without unnecessary margins, by tabulating accurately each component of the cooling load.
5. It permits the engineer to select the type of control necessary for providing flexibility to meet all conditions of operation by indicating the load components and the manner in which they vary.

The manner in which this method provides the benefits summarized will become apparent in the course of the description.

Components of Heat Gain

The cooling load is composed of five different components:

1. Heat conducted through walls, windows, etc.
2. Heat absorbed from radiations of the sun.
3. Heat generated by lights and appliances, and other miscellaneous sources.
4. Heat brought in by outdoor air.
5. Heat liberated by people.

The components of heat gain, classified by source, may be further classified as sensible and latent heat gain. The first two components fall into the classification of sensible heat gain; that is, they tend to raise the temperature of the air within the structure. The last three components also produce sensible heat gain, but in addition they may produce latent heat gain; that is, they may tend to increase the moisture content of the air within the structure. Each component is discussed briefly below.

Heat is conducted through walls and partitions because the temperature of the air within the air conditioned space is lower than that on the opposite side of the wall. This is the same process as that by which heat is lost through walls in winter.

Heat absorbed from radiations of the sun increases the cooling load in two ways. In the first place, the sun transmits invisible, but intense, heat rays which increase the temperature of all surfaces exposed to them. With this higher temperature, more heat is conducted to the interior of the structure than if the walls were not exposed to the sunshine. In the second place, the heat rays of the sun pass almost undiminished through ordinary window glass which is exposed to them.

Temperatures of flat roofs have been observed to rise as high as 180 F on a bright summer day, 75 to 80 deg above the temperature of the air. The heat conduction into top floor rooms may be more than doubled because of this effect. Similarly, glass windows exposed to the sun may allow as much as 90 per cent of the incident heat to pass through, and it is not uncommon for this portion of the cooling load to be more than the total conduction through the wall proper.

Heat is generated by energy consuming appliances within the air conditioned space, such as electric lights, motors, coffee urns, steam tables, etc. Some of the appliances, such as electric lights and motors, produce only sensible heat gain. Other appliances, such as coffee urns, produce both sensible and latent heat gains. Latent heat gains may arise from the evaporation of moisture within the appliance, or from the liberation of moisture as a product of combustion where a gas flame is present.

Heat gain results from the incoming outdoor air, which may be introduced by natural, uncontrolled infiltration or by controlled ventilation. This is both

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sensible and latent in character, as outdoor air entering the structure may have to be both cooled and dehumidified.

Heat is liberated by people through convection and radiation from the surface of the body, and the evaporation of moisture from the skin. The former adds to the sensible cooling load, and the latter to the latent cooling load. Where a number of people are in a relatively small space, the heat they contribute becomes an important item. The rate at which heat is contributed by a normal person not engaged in undue activity may be compared to the absorption of an average domestic refrigerator, or to that required to completely boil away a pint of water in 3 hours.

Relation Between Total Heat Gain and Individual Components

A complete heat gain calculation to determine the maximum requirements for cooling involves not only the independent calculation of each of the various components, but it involves also the proper combination of these components to determine the correct maximum heat gain. Each individual component varies in magnitude from hour to hour during the day and reaches a maximum or peak value at some particular time. However, the individual maximums do not necessarily occur simultaneously, so that the maximum total heat gain actually may be less than the sum of the maximum values of the several components. For example, refer to Fig. 1, which shows a family of curves worked up in the calculation of heat gain for a group of offices.

Curves 1 and 2 show the heat conducted through walls, and the sensible heat of ventilation air, respectively. The variations in these components result from changes in the temperature of the outdoor air.

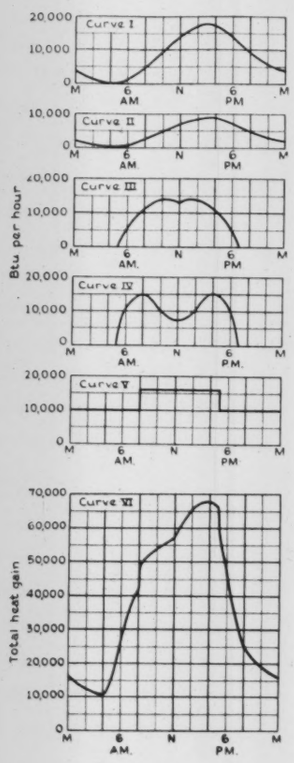


Fig. 1—Time variations of components and total of cooling load in a suite of offices

Curve I—Conduction from outdoor-indoor temperature difference

Curve II—Sensible heat from ventilation air

Curve III—Additional conduction from sun effect

Curve IV—Sun effect through windows

Curve V—Latent heat of ventilation air and heat from people

Curve VI—Total heat gain

Curve 3 shows the variation in additional conduction through walls exposed to the sun. The intensity of the solar heat striking a wall depends on the angle between the wall and the rays of the sun. This intensity is greater when the rays of the sun are most nearly perpendicular to the wall, and hence the progression of the sun from east to west causes a variation in this component of the heat gain. Similarly, curve 4 shows the variation in heat gain from solar heat passing directly through windows.

Curve 5 shows the load imposed by the latent heat gain of ventilation air, with the heat from occupants superposed during the period of day when such occupants are present. The reason why the latent heat of ventilation air shows no variation will be discussed later.

Curve 6 shows the total heat gain, which is obtained by adding each of the components in the proper phase relation. An analysis of Fig. 1 shows that whereas individual components attain maximum values at 8 a.m., 10 a.m., 2 p.m., 3 p.m., and 4 p.m., respectively, the total heat gain is a maximum at 4 p.m. Clearly it would not be accurate (though it would be conservative) to find the greatest value that each component attains during the day, and then to add all the maximum values together to find the maximum total gain. The difference between the results obtained by adding the maximum values of the individual components, and combining these components at the time of day when the total is maximum, may be as much as 20 per cent, or as little as 5 per cent. The method of calculation described here is sufficiently flexible to permit taking account of the fact that when one component is at its peak, others may have either passed or not yet reached their peak values. The introduction of this element of additional accuracy and flexibility requires that the calculations be made for a predetermined time of day in each case. The method of predicting the correct time of day will be described later.

Design Conditions

The following design conditions used in making a heat gain calculation affect the magnitude of the cooling load:

1. Outdoor air
   (a) Dry-bulb temperature
   (b) Vapor density
2. Indoor air
   (a) Dry-bulb temperature
   (b) Vapor density
3. Weight of outdoor air entering the enclosure.

The conditions of the indoor air are fixed by the dry-bulb and wet-bulb temperatures which are specified according to the requirements of the application.

The amount of ventilation air is specified according to the requirements of the application if ventilation is controlled. Otherwise it is determined by the infiltration rate.

Design outdoor conditions are based on the Weather Bureau records of the locality under consideration. The design outdoor dry-bulb temperature is usually specified as a maximum value. This design temperature for comfort applications may be selected so that it will be exceeded by not more than about 90 per cent of the

daily maximum outdoor temperatures during the cooling seasons over a period of several years. Systems designed on the basis of such an outdoor temperature will, on occasional extreme days, be incapable of maintaining normal indoor conditions, but will be generally satisfactory, considering both performance and investment. There are several references [1], [2] on the subject of design outdoor conditions, and recommended values are given in Chapter 8 of THE A.S.H.V.E. Guide[3].

The design outdoor temperature described is really the design maximum outdoor temperature. As has already been described, the maximum total heat gain may occur at some time of day when the outdoor temperature is considerably lower than its maximum value. A night club is an outstanding example of an application in which this condition arises. Hence, it is necessary to know not only the design maximum outdoor temperature, but also how the outdoor temperature varies from hour to hour, and how far it may have dropped by evening following a *maximum* day. A careful study of Weather Bureau data indicates that for practical purposes the outdoor temperature may be assumed to vary in the manner shown by Fig. 2. It has been found that on the average the outdoor temperature is highest about 3 p.m., and lowest about 5 a.m. The difference between the maximum and minimum outdoor temperatures on a *maximum* day may be taken as the mean difference between the daily maximum and daily minimum outdoor temperatures for the month of July. This mean difference is termed by the Weather Bureau the *mean daily range* for July. It varies from about 10 F in some parts of the United States to about 40 F in other localities. A table of outdoor temperatures for various hours of the day may then be constructed by using the formula:

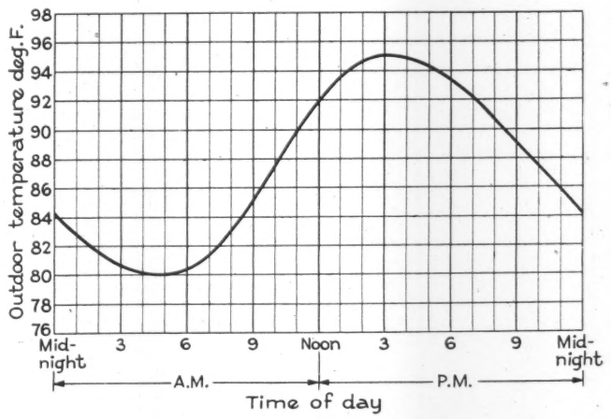


Fig. 2—Typical variation of outdoor temperature

$$t_o = t_d - B\,V \qquad (1)$$

where

t_o = design outdoor temperature at a particular time of day.

t_d = design maximum outdoor temperature.

V = mean daily range for July.

B = an hourly factor, determined by the shape of the curve in Fig. 2, and tabulated in Table 1.

[1] The Relation of Climate to Air Conditioning Design, by O. W. Armspach, *Heating and Ventilating*, Vol. 29, Dec. 1932, p. 26.

[2] Climate in Air Conditioning, by H. W. Skinner, *Heating and Ventilating*, Vol. 29, Dec. 1932, p. 23.

[3] AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS GUIDE, 193[illegible]

Table 1—Design Outdoor Temperatures for Various Hours of the Day

(t_d = 95 F; V = 15 F)

Hourly Factor B	BV	Design Outdoor Temperature ($t_o = t_d - BV$)	Time of Day
0.8	12.0	83	1 a.m.
0.9	13.5	82	2
0.95	14.2	81	3
1.0	15.0	80	4
1.0	15.0	80	5
1.0	15.0	80	6
0.9	14.2	81	7
0.8	12.0	83	8
0.65	9.8	85	9
0.5	7.5	88	10
0.3	4.5	91	11
0.2	3.0	92	Noon
0.1	1.5	94	1 p.m.
0	0	95	2
0	0	95	3
0	0	95	4
0.05	0.75	94	5
0.1	1.5	94	6
0.2	3.0	92	7
0.3	4.5	91	8
0.4	6.0	89	9
0.5	7.5	88	10
0.6	9.0	86	11
0.7	10.5	85	Midnight

Table 1 shows design outdoor temperatures, calculated as outlined above, for various hours of the day in a locality where the design maximum outdoor temperature is 95 F and the mean daily range for July is 15 F. Of course there are occasional days when the temperature range is less than the mean value, and the temperature is somewhat higher in the evening than is indicated by the above analysis. This possibility should be kept in mind.

It was implied previously that the latent heat gain of ventilation air is substantially constant for various hours of the day. A study of Weather Bureau Data shows that in general the outdoor relative humidity is lowest when the outdoor dry-bulb temperature is highest, and vice versa. Furthermore, this variation in outdoor relative humidity is of such a nature that the vapor density, or moisture content, of outdoor air is approximately constant. Consequently, the outdoor vapor density may be determined from the psychrometric chart once the design maximum outdoor dry-bulb and wet-bulb temperatures are known. The design outdoor wet-bulb temperature may be selected on the same basis as the design outdoor dry-bulb temperature; that is, high enough so that it will include about 90 per cent of the daily maximum wet-bulb temperatures during the cooling season.

The outdoor vapor density is not exactly constant during the entire day, and of course shows sharp variations during stormy periods. The assumption of a constant design outdoor vapor density on a *maximum* day, however, is reasonably accurate for comfort applications. It must be recognized that all assumptions in regard to climatic conditions are of necessity approximate.

Time Lag Resulting from Heat Storage

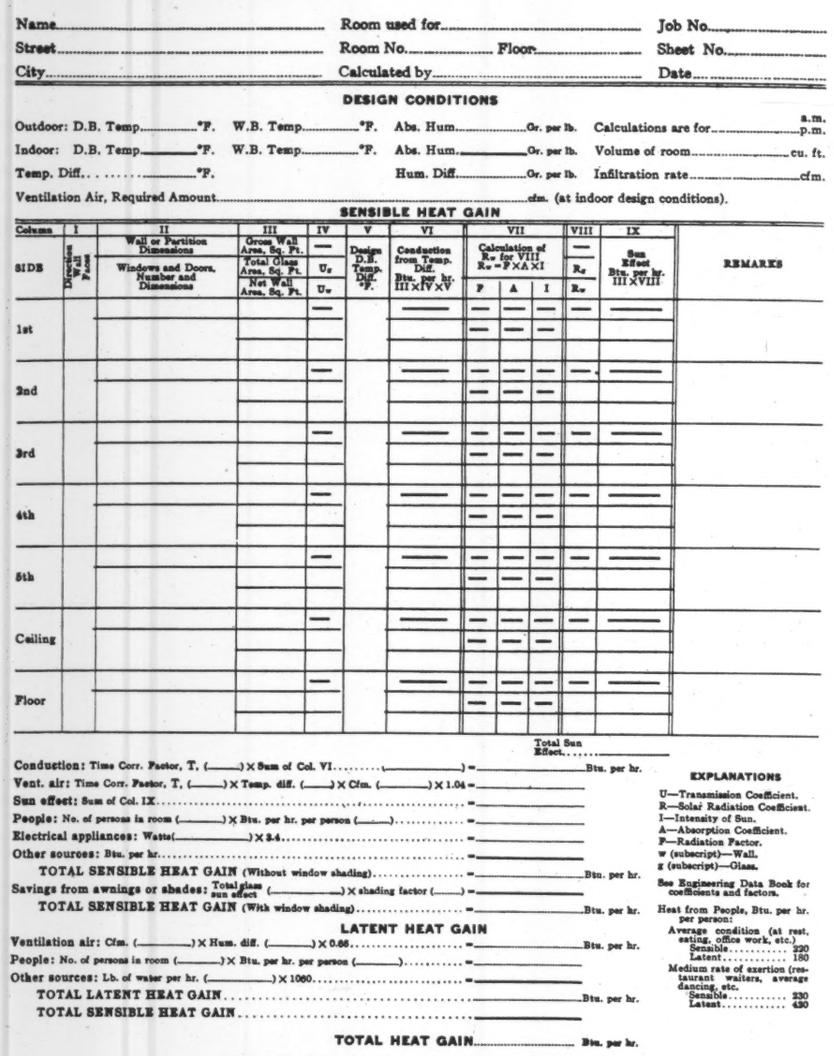
The heat storage capacity of walls results in a phenomenon akin to a time lag in the flow of heat through them. The result is that the maximum effects of outdoor temperature and solar radiation may not manifest themselves on the interior until some time after the actual maximum outdoor conditions have passed. This introduces the problem of determining time lag, which will be discussed later.

Summary of Analysis

The preceding paragraphs have discussed briefly the five components of the heat gain which were summarized in the beginning, and further classified them as sensible and latent. These components have been shown to vary during the day in a manner dependent on climatic and indoor conditions. This causes a phase relation between them of such a nature that the correct maximum heat gain is not necessarily the sum of the maximum values of the components, but is the sum taken at the particular time of day when the total is a peak. This is further affected by time lag in walls.

Design conditions which affect the character and magnitude of the heat gain have been listed, and a method suggested for determining design outdoor conditions. The determination of design indoor conditions has been omitted because they are usually specified by the requirements of the application, and depend on a number of factors which are often beyond the control of one who is making the heat gain calculation.

The following sections of this paper will be devoted to methods of determining accurately the character and magnitude of the components of the heat gain, and of combining them properly to predict the total cooling load.



Name ____ Room used for ____ Job No. ____
Street ____ Room No. ____ Floor ____ Sheet No. ____
City ____ Calculated by ____ Date ____

DESIGN CONDITIONS

Outdoor: D.B. Temp. ____ °F. W.B. Temp. ____ °F. Abs. Hum. ____ Gr. per lb. Calculations are for ____ a.m. p.m.
Indoor: D.B. Temp. ____ °F. W.B. Temp. ____ °F. Abs. Hum. ____ Gr. per lb. Volume of room ____ cu. ft.
Temp. Diff. ____ °F. Hum. Diff. ____ Gr. per lb. Infiltration rate ____ cfm.
Ventilation Air, Required Amount ____ cfm. (at indoor design conditions).

SENSIBLE HEAT GAIN

Column	I	II	III	IV	V	VI	VII			VIII	IX	
SIDE	Direction Wall Faces	Wall or Partition Dimensions / Windows and Doors, Number and Dimensions	Gross Wall Area, Sq. Ft. / Total Glass Area, Sq. Ft. / Net Wall Area, Sq. Ft.	U_c / U_w	Design D.B. Temp. Diff. °F.	Conduction from Temp. Dif. Btu. per hr. III × IV × V	Calculation of R_w for VIII $R_w = F \times A \times I$			R_w	Sun Effect Btu. per hr. III × VIII	REMARKS
							F	A	I			
1st												
2nd												
3rd												
4th												
5th												
Ceiling												
Floor												
									Total Sun Effect			

Conduction: Time Corr. Factor, T, (____) × Sum of Col. VI (____) = ____ Btu. per hr.
Vent. air: Time Corr. Factor, T, (____) × Temp. dif. (____) × Cfm. (____) × 1.04 = ____
Sun effect: Sum of Col. IX = ____
People: No. of persons in room (____) × Btu. per hr. per person (____) = ____
Electrical appliances: Watts (____) × 3.4 = ____
Other sources: Btu. per hr. = ____
TOTAL SENSIBLE HEAT GAIN (Without window shading) = ____ Btu. per hr.
Savings from awnings or shades: Total glass sun effect (____) × shading factor (____) = ____
TOTAL SENSIBLE HEAT GAIN (With window shading) = ____ Btu. per hr.

LATENT HEAT GAIN

Ventilation air: Cfm. (____) × Hum. diff. (____) × 0.68 = ____ Btu. per hr.
People: No. of persons in room (____) × Btu. per hr. per person (____) = ____
Other sources: Lb. of water per hr. (____) × 1060 = ____
TOTAL LATENT HEAT GAIN = ____ Btu. per hr.
TOTAL SENSIBLE HEAT GAIN = ____
TOTAL HEAT GAIN = ____ Btu. per hr.

EXPLANATIONS

U—Transmission Coefficient.
R—Solar Radiation Coefficient.
I—Intensity of Sun.
A—Absorption Coefficient.
F—Radiation Factor.
w (subscript)—Wall.
g (subscript)—Glass.

See Engineering Data Book for coefficients and factors.

Heat from People, Btu. per hr. per person:
Average condition (at rest, eating, office work, etc.)
Sensible 220
Latent 180
Medium rate of exertion (restaurant waiters, average dancing, etc.
Sensible 230
Latent 430

Fig. 3—Heat gain calculation sheet

Calculation of the Components of Heat Gain

Heat Gain Calculation Sheet

A calculation sheet for organizing and tabulating the calculations described in the following paragraphs is shown in Fig. 3.

Conduction From Temperature Differences

The technique of calculating heat conduction through a wall is well known and widely used in determining heating requirements for winter. The heat gain by conduction is calculated from the formula:

$$H_c = A_w U_w (t_o - t_i) \quad (2)$$

where

H_c = heat conducted through the wall, window or other partition, Btu per hour.
A_w = Area of wall, window, etc., sq ft
t_i = design indoor temperature, F
t_o = design outdoor temperature, F
U_w = overall coefficient of heat transmission of the wall, window, etc., Btu per hour per sq ft per F.

A complete discussion of the transmission of heat through walls, etc., is given in THE A.S.H.V.E. GUIDE[3], Chapter 5.

Table 2—Transmission Coefficients, Temperature Differences and Solar Radiation Coefficients for Ceilings

Location of Room	U_w	Temp. Diff.	R_w
Not on top floor	$U_w = U_c$	$(t_o - t_i)$ — (3 to 5 Deg)	$R_w = 0$
Under unventilated attic with sloping roofs (ordinary house construction)	$U_w = \dfrac{U_c}{1 + 0.7 \dfrac{U_c}{U_r}}$	$(t_o - t_i)$	$R_w = FaI$
Under unventilated attic with flat roof	$U_w = \dfrac{U_c}{1 + \dfrac{U_c}{U_r}}$	$(t_o - t_i)$	$R_w = FaI$
Under well ventilated attic (natural ventilation)	$U_w = U_c$	(Attic Temp. — t_i)	$R_w = 0$
Under fan ventilated attic	$U_w = U_c$	$(t_o - t_i)$ + (5 to 10 Deg)	$R_w = 0$
Directly under flat roof (roof also forms the ceiling)	$U_w = U_r$	$(t_o - t_i)$	$R_w = FaI$
Directly under sloping roof (roof also forms the ceiling)	$U_w = \dfrac{\text{Roof area}}{\text{Horizontal Ceiling Area}} U_r$	$(t_o - t_i)$	$R_w = FaI$

Notes:
1. t_o = Design outdoor dry-bulb temperature.
2. t_i = Design indoor dry-bulb temperature.
3. U_w = Overall transmission coefficient.
4. U_c = Transmission coefficient for ceiling alone.
5. U_r = Transmission coefficient for roof alone.
6. F = Radiation factor. Determine F to use in fourth column from Fig. 5 by using the value of U_w calculated in the second column.
7. a = Absorption coefficient. (See table 4.)
8. I = Intensity of sun. Determine I to use in fourth column from Table 3, using values for horizontal surfaces.

together with transmission coefficients for walls of various types of building construction. Special cases, which arise in connection with rooms under attics and sloping roofs, are covered in Table 2 of this paper.

In making this calculation, care must be exercised to consider separately each section of the wall having different overall heat transmission coefficients or different temperature differences. For example, the overall coefficient for a window is different from that for the wall in which it is located. This is taken care of automatically on the calculation sheet.

Careful judgment must be exercised in selecting temperature differences. Thus, the value for outside walls is equal to the difference between the design outdoor temperature and the design indoor temperature, while the temperature difference across inside partitions may be greater or less. For inside partitions, with no unusual sources of heat on the far side, it is justifiable to assume temperature differences from 3 to 5 deg less than for outside walls, at that time of day when the outdoor temperature is highest. If the partition adjoins a hot kitchen, or similar heated space, it is desirable to assume a temperature difference somewhat greater than for the outside walls.

Sun Effect Through Walls

The heat absorbed from solar radiation falling on exposed surfaces has already been discussed briefly. The actual process that occurs can be explained simply in connection with Fig. 4. The amount of heat received by each square foot of the wall surface depends upon the intensity of the solar heat and the angle between the surface and the rays of the sun. However, a part of this heat is reflected directly back into space without affecting the temperature of the wall surface, because all surfaces act as heat mirrors to a greater or lesser extent. The nature of the outside surface of the wall determines its effectiveness as a heat mirror, and thus is an important factor in determining how much of the incident solar heat penetrates the wall and adds to the cooling load. Of the heat absorbed by the outside surface of the wall, a portion is dissipated to the outside air and surroundings by convection and radiation and the balance is conducted through the wall into the air conditioned space.

The actual *total* amount of heat conducted through a wall exposed to the sun depends not only on the amount of solar heat which is absorbed by the outside surface but also on the overall heat transmission coefficient, and the temperatures of the indoor and outdoor air. A mathematical expression for the heat balance involved shows that the total heat conducted to the interior may, for practical purposes, be expressed by,

$$H_t = A_w\ U_w\ (t_o - t_i) + A_w\ R_w \qquad (3)$$

The first term of (3) is identical with (2). The second term is

$$H_r = A_w\ R_w \qquad (4)$$

where

H_r = Additional heat conducted through a wall exposed to the sun, Btu per hour

$R_w = F\ a\ I$ (5)

I = Actual solar radiation striking the wall, Btu per hour per sq ft

a = Percentage (expressed as a decimal) of the incident solar radiation which is absorbed by the wall surface.

F = Percentage (expressed as a decimal) of the absorbed solar radiation which is transmitted to the inside.

It will be noted in (3) that the total conduction through a wall includes two additive terms. The first one is identically the same as that given above for heat conducted through a wall not exposed to the sun. The second term depends only on the intensity of the sun and the characteristics and orientation of the wall, and is independent of the temperature difference across the wall. Thus, equation (4) gives the *additional conduction* through walls exposed to the sun.

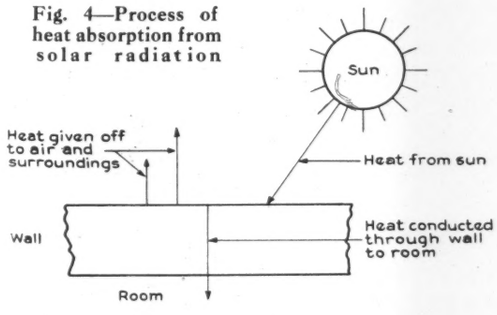
The radiation factor R_w, depends on I, the amount of solar heat incident on the wall; a, the absorption coefficient of the wall; and F, a factor which gives the percentage of the absorbed solar heat which is transmitted to the interior.

The intensity of solar radiation, I, on a given wall, depends on the amount of water vapor and dust in the atmosphere[4] through which the solar heat must travel before it reaches the surface, and the angle between the rays of the sun and the surface[5], [6]. Table 3 gives values of I for surfaces facing different directions, at different hours of the day and for several northern latitudes. These values of I take into account the average amount of water vapor and dust in the atmosphere throughout the United States, and are applicable for the period of the year during which the heat gain is normally a maximum, namely, from early May to the middle of August.

Average values of the absorption coefficient, a, are given in Table 4. The factor F depends on the overall transmission coefficient, U_w of the wall. Values of this factor are determined from the curve of Fig. 5.

It should hardly be necessary to point out here that the calculations of sun effect through walls described above are made only when the walls are definitely exposed to the sun. If the wall is shaded by trees or other buildings, there will be no heat from the sun striking it directly, and the calculation of this component of the heat gain is omitted. When the wall is only partly shaded, it will be necessary to estimate, or calculate from the geometry involved[4] what percentage of the wall is thus protected, and to make allowances accordingly.

Fig. 4—Process of heat absorption from solar radiation



[4]The Determination of Sun Effect on Summer Cooling Loads, by Hendrickson and Walker, *Heating and Ventilating*, Vol. 30, June 1933, p. 20.

[5]Summer Cooling for Comfort as Affected by Solar Radiation, by Hendrickson and Walker, *Heating and Ventilating*, Vol. 29, Nov. 1932, p. 14.

[6]Studies of Solar Radiation Through Bare and Shaded Windows, by Houghten, Gutberlet and Blackshaw, A.S.H.V.E. JOURNAL, *Heating, Piping and Air Conditioning*, Vol. 6, Feb. 1934, p. 67.

Table 3—Sun Effect Coefficients at Various Latitudes for Walls Facing Several Directions

I = Btu per hour striking 1 sq ft of wall surface.

R_g = Btu per hour transmitted by 1 sq ft of single glass (for double glass multiply values of R_g given below by 0.9).

North Latitude, Degrees	Coeff.	Time of Day	Direction Wall Faces Ne	E	Se	S	Sw	W	Nw	Horizontal Surface
20	I, for walls and roofs	6 a.m.	100	105	50					15
		7	165	185	95					70
		8	175	205	115					140
		9	150	185	110					205
		10	105	135	90					260
		11	55	70	50					290
		Noon								300
		1 p.m.					50	70	55	290
		2					90	135	105	260
		3					110	185	150	205
		4					115	205	175	140
		5					95	185	165	70
		6					50	105	100	15
	R_g, for windows and skylights	6 a.m.	95	100	40					
		7	155	170	80					50
		8	155	190	95					120
		9	125	165	85					185
		10	75	110	60					235
		11	25	40	20					265
		Noon								275
		1 p.m.					20	40	25	265
		2					60	110	75	235
		3					85	165	125	185
		4					95	190	155	120
		5					80	170	155	50
		6					40	100	95	
25	I, for walls and roofs	6 a.m.	115	120	60					20
		7	165	190	105					75
		8	165	210	125					145
		9	135	185	125					210
		10	90	135	105	10				260
		11	35	70	65	25				290
		Noon			20	30	20			300
		1 p.m.				25	65	70	35	290
		2				10	105	135	90	260
		3					125	185	135	210
		4					125	210	165	145
		5					105	190	165	75
		6					60	120	115	20
	R_g, for windows and skylights	6 a.m.	105	115	45					
		7	150	175	85					55
		8	150	190	105					125
		9	110	165	100					190
		10	60	110	75					235
		11	10	40	35					265
		Noon								275
		1 p.m.					35	40	10	265
		2					75	110	60	235
		3					100	165	110	190
		4					105	190	150	125
		5					85	175	150	55
		6					45	115	105	
30	I, for walls and roofs	6 a.m.	125	135	65					25
		7	165	195	110					80
		8	160	210	135					150
		9	125	185	140	10				210
		10	75	135	120	30				255
		11	20	70	85	45				285
		Noon			35	50	35			295
		1 p.m.				45	85	70	20	285
		2				30	120	135	75	255
		3				10	140	185	125	210
		4					135	210	160	150
		5					110	195	165	80
		6					65	135	125	25
	R_g, for windows and skylights	6 a.m.	115	125	50					10
		7	150	180	95					60
		8	140	190	115					125
		9	95	165	110					190
		10	45	110	90	10				235
		11		40	55	20				260
		Noon			10	20	10			270
		1 p.m.				20	55	40		260
		2				10	90	110	45	235
		3					110	165	95	190
		4					115	190	140	125
		5					95	180	150	60
		6					50	125	115	10
35	I, for walls and roofs	6 a.m.	135	145	70					30
		7	160	200	120					85
		8	150	210	145					150
		9	110	185	150	30				205
		10	60	135	135	55				250
		11		70	100	70				280
		Noon			55	75	55			290
		1				70	100	70		280
		2				55	135	135	60	250
		3				30	150	185	110	205
		4					145	210	150	150
		5					120	200	160	85
		6					70	145	135	30
	R_g, for windows and skylights	6 a.m.	120	135	55					15
		7	145	185	100					65
		8	130	190	125					130
		9	85	165	125					185
		10	30	110	105	25				230
		11		40	70	40				255
		Noon			25	45	25			265
		1 p.m.				40	70	40		255
		2				25	105	110	30	230
		3					125	165	85	185
		4					125	190	130	130
		5					100	185	145	65
		6					55	135	120	15
40	I, for walls and roofs	6 a.m.	140	155	80					35
		7	160	200	125					90
		8	140	210	155	10				150
		9	100	185	160	45				200
		10	40	135	150	75				240
		11		70	115	95	15			270
		Noon			70	100	70			280
		1 p.m.			15	95	115	70		270
		2				75	150	135	40	240
		3				45	160	185	100	200
		4				10	155	210	140	150
		5					125	200	160	90
		6					80	155	140	35
	R_g, for windows and skylights	6 a.m.	130	140	65					20
		7	145	185	105					70
		8	120	190	135					130
		9	70	165	140	20				180
		10	15	110	125	45				220
		11		40	90	65				250
		Noon			40	70	40			255
		1 p.m.				65	90	40		250
		2				45	125	110	15	220
		3				20	140	165	70	180
		4					135	190	120	130
		5					105	185	145	70
		6					65	140	130	20
45	I, for walls and roofs	5 a.m.	70	70	25					
		6	145	165	85					40
		7	155	205	130					95
		8	130	210	165	25				145
		9	85	180	175	65				195
		10	25	135	160	95				230
		11		70	130	115	30			260
		Noon			90	125	90			265
		1 p.m.			30	115	130	70		260
		2				95	160	135	25	230
		3				65	175	180	85	195
		4				25	165	210	130	145
		5					130	205	155	95
		6					85	165	145	40
		7					25	70	70	
	R_g, for windows and skylights	5 a.m.	65	65	20					
		6	135	150	70					25
		7	140	190	115					70
		8	110	190	145					125
		9	55	165	150	35				175
		10		110	140	65				215
		11		40	105	85	10			240
		Noon			55	95	55			245
		1 p.m.			10	85	105	40		240
		2				65	140	110		215
		3				35	150	165	55	175
		4					145	190	110	125
		5					115	190	140	70
		6					70	150	135	35
		7					20	65	65	

Certain special cases arise in connection with rooms located under attics, or under flat or sloping roofs. Table 2 gives instructions for determining R_w to use in such cases. These are based on using the horizontal ceiling area to determine total sun effect through the ceiling.

The method of calculating sun effect through walls described in this paper makes possible an accurate and scientific determination of this component. Fundamentally, this method is equivalent to calculating the temperature differences across walls exposed to the sun, and avoids the necessity for estimating them. The mathematical elimination of the wall surface temperature, however, results in the simple expressions given by equations (3), (4), and (5). Furthermore, when the above method is used in conjunction with the proper combination of the several components, sun effect is included for only those walls exposed to the sun when the total cooling load is a maximum.

Table 4—Solar Absorption Coefficients (a) for Different Building Materials

Surface Material	Absorption Coefficient (a)
VERY LIGHT COLORED SURFACES	
Such as	
White stone	0.4
Very light colored cement	
White or light cream-colored paint	
MEDIUM DARK SURFACES	
Such as	
Asbestos shingles	0.7
Unpainted wood	
Brown stone	
Brick and red tile	
Dark-colored cement	
Stucco	
Red, green or gray paint	
VERY DARK COLORED SURFACES	
Such as	
Slate roofing	0.9
Tar roofing materials	
Very dark paints	

Sun Effect Through Windows

Transparent windows present a problem somewhat different from that of opaque walls, because they permit a large percentage of the solar energy to pass through undiminished. The actual amount of incident energy which is transmitted depends on the actual transparency of the window glass to the solar heat rays and on the angle between the rays and the surface of the glass[5], [6]. That is, the amount of heat reflected by the exterior surface of the glass depends upon the angle between the glass and the rays of the sun. An ordinary window glass which is exactly perpendicular to the rays of the sun actually permits about 90 per cent of the energy to pass through unobstructed[7], [8]. The heat gain through windows exposed to the sun is calculated by the formula:

$$H'_r = A_g R_g \qquad (6)$$

where

H'_r = solar radiation transmitted through a window, Btu per hour

A_g = Area of the glass, sq ft

R_g = amount of solar heat transmitted directly, through the glass, Btu per hour per sq ft.

Values of the radiation coefficient, R_g, for windows in walls facing different directions, at different hours of the day and for several northern latitudes are given in Table 3. These values of R_g are based on the values of I given in the same table and the reflective and absorptive characteristics of ordinary window glass at various angles between the solar rays and the glass.

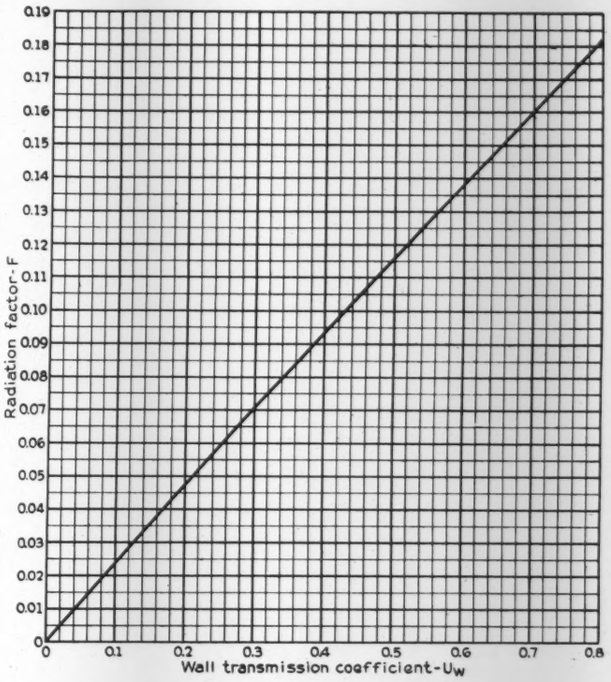


Fig. 5—Solar radiation factor vs. overall wall transmission coefficient

Recently there have been developed several special kinds of glass which are more absorptive to solar heat than ordinary window glass[9]. If the structure is glazed with windows of this character, the values of R_g given in Table 3 should be reduced in varying percentages depending on the nature of the glass used.

When windows are completely shaded from the sun by trees or other buildings, then none of the solar energy passes through them. If they are partially shaded the percentage reduction must be estimated, or calculated from the geometry involved. This is discussed by Hendrickson and Walker[4]. In the absence of such accidental shading, there are various man-made devices such as awnings, shades, and venetian blinds which may be used to mitigate the effect of the sun to a greater or lesser extent. It is somewhat difficult to give an exact appraisal of the relative effectiveness of these various devices, because as yet complete experimental evidence is not available. However, the matter may be summarized about as follows[6]:

1. Devices hung *inside* the window, such as shades or blinds, are not as effective as devices hung outside. This is because they interrupt the rays of the sun only after they have passed through the glass. Although some of the heat may be reflected back through the glass, a portion of it is absorbed and transmitted into the room.
2. Awnings are probably not so effective on the first floor of a building as they are on upper floors. In the former case the surface of the ground no doubt reflects some of the solar heat through the opening under the awning, and there is secondary radiation from the warm awning itself.

[5] Surface Absorption of Heat from Solar Radiation, by Hechler and Que[illegible], *Refrigerating Engineering*, Vol. 25, Feb. 1933, p. 86.

[6] Radiation of Energy Through Glass, by Blackshaw and Houghten, A.S[illegible] V.E. Journal, *Heating, Piping and Air Conditioning*, Oct. 1933, p. 5[illegible]

[9] Windows—and Their Relation to Air Conditioning Problems, by W. W. Shaver, *Refrigerating Engineering*, Vol. 26, Sept. 1933, p. 133.

3. Awnings on the first floor may be assumed to eliminate from the heat gain approximately 75 per cent of the solar radiation which would otherwise pass through the bare window, and awnings on upper floors may be assumed to exclude 85 per cent of the solar radiation from the window.
4. Inside shades, buff-colored, clean, and completely drawn may be expected to eliminate about 50 per cent of the solar radiation which would otherwise pass through the window Dark shades are not nearly so effective.
5. Inside venetian blinds with metallic aluminum surfaces, are about as effective as inside buff shades.

Heat from Lights and Appliances

Heat dissipating appliances which give off only sensible heat may be divided into three classes:

1. Those whose electrical watts input are known, or can be read from a nameplate.
2. Those whose electrical watts input may be determined, through brief calculations.
3. Those which have no electrical input.

In the first category may be found such devices as electric lights, toasters, waffle irons, etc., for which the nameplate indicates the electrical power consumption. All of this power is changed into heat and given off to the air in the room. The amount of heat in Btu per hour thus liberated is equal to 3.4 times the watts input (one watt is equivalent to 3.415 Btu per hour).

Under the second heading may be included such devices as motors which show on their nameplate a hp rating but do not give the watts input for that rating. If the motor efficiency is known, the watts input may be calculated from the formula:

$$P = \frac{746\ (hp)}{n} \qquad (7)$$

where

P = motor input, watts.
(hp) = motor load, horsepower.
n = motor efficiency (expressed as a decimal).

If there is no way of determining the motor efficiency, then as a fair approximation, the heat gain from such sources may be obtained by using the data in Table 5, multiplying the figures in the table by the motor hp rating.

The third class of appliances includes items such as ranges, steam tables, coffee urns, uncovered steam pipes, etc., where no electrical power consumption is involved. For such cases it is necessary to make a direct estimate of the heat gain in Btu per hour. Table 6 gives heat from various types of appliances.

There are some appliances which give off not only sensible heat, but latent heat as well. These will be encountered most generally in industrial applications. Any device which evaporates moisture into the air falls into this category, and the moisture thus evaporated must be included in the calculations. First, calculate the amount of moisture which is evaporated. The latent heat gain from such appliances is then:

$$H_l = 1060\ M$$

where

H_l = latent heat of appliance, Btu per hour.
M = rate of evaporation of water, lb per hour.
1060 = average latent heat of vaporization of water vapor at air conditioning temperatures.

A pitfall to avoid in this calculation is the inclusion of latent heat from appliances in two different parts of the calculation. Thus, suppose an electric hot plate rated at 500 watts were used to boil water, resulting in the evaporation of a half pound an hour. The total heat dissipated by the hot plate would be $500 \times 3.4 = 1700$ Btu per hour. The latent heat given off would be $0.5 \times 1060 = 530$ Btu per hour. The thing to note is that the 530 Btu is included in the 1700 Btu given off by the hot plate. Thus,

Latent heat from the hot plate = 530 Btu per hour
Sensible heat from hot plate = 1170 Btu per hour
Total heat from hot plate = 1700 Btu per hour

Table 5—Heat Generated by Motors and Motor Generators Operating Continuously

Nameplate Rating in HP	Heat Gain in Btu per hr per hp Rating: Connected Load in Same Room	Connected Load Outside of Room
⅛ to ½	4250	1700
½ to 3	3700	1150
3 to 20	2950	400

Motor Generators

Nameplate Rating in Kilowatts	Heat Gain in Btu per hr per kw Rating
¾ to 3	2800
3 to 5	1300

Table 6—Heat from Appliances

Nature of Appliance	Btu per hr Not Hooded	Hooded and Ventilated
Coffee Urns and Percolators		
1 gallon capacity	1000	250
2	2000	500
3	3000	750
4	4000	1000
5	5000	1250
10	16000	4000
Dish Warmers per 10 Sq ft of Shelf	6000	
Restaurant Range—4 Burner and Oven	100000	
Residence Electric Range		
Small Burner	3400	
Medium Burner	4100	
Large Burner	7700	
Oven	10200	
Appliance Connection	2200	
Warming Compartment	1000	
Residence Gas Range		
Giant Burner	12000	
Medium Burner	9000	
Pilot	250	
Oven	1000 Btu per cu ft of oven volume	

Heat from Ventilation Air

Outdoor air may be introduced to an air conditioned space either by natural and uncontrolled infiltration or by controlled ventilation. Such ventilation air must be cooled to room temperature and dehumidified to room humidity. The sensible and latent heat gains from ventilation air are determined accurately from the two formulas:

$$H_s = \frac{60\ Q}{v}\ 0.24\ (t_o - t_i) \qquad (10)$$

$$H_l = \frac{60\ Q}{v}\ \frac{\omega_o - \omega_i}{7000}\ 1060 \qquad (11)$$

where

H_s = sensible heat, Btu per hour.
H_l = latent heat, Btu per hour.
Q = ventilation air rate, cfm
v = specific volume of ventilation air, cu ft per lb of dry air.
ω_o = vapor density of outdoor air, grains of moisture per lb of dry air.
ω_i = vapor density of indoor air.

For all practical purposes, these formulas can be simplified to:

$$H_s = 1.04\ Q\ (t_o - t_i) \quad (10a)$$
$$H_l = 0.66\ Q\ (\omega_o - \omega_i) \quad (11a)$$

The various terms involved in these two formulas are easily determined. The method of determining the outdoor and indoor dry-bulb temperatures and vapor densities have been described previously under Design Conditions. The quantity of ventilation air, Q, is equal to the specified amount of ventilation air if a ventilation duct leads to the air conditioner. If dependence is placed on natural infiltration to supply outdoor air, Q is the infiltration rate.

Heat from People

The heat given off by persons varies with the degree of activity of the individual and the temperature of the air, as described in Chapter 2 of THE A.S.H.V.E. GUIDE[3]. In most air conditioning applications people are found in one of two general states of activity—either they are seated or standing at rest, moving about only occasionally (as in a home or office); or they are engaged in brisk activity such as walking or dancing (for example, restaurant waiters or night club patrons). For all ordinary cases, the values of sensible and latent heat gain in Btu per hour per person as given in Table 7 may be used. Consideration should, of course, be given in all cases to the number of people who fall in one category or another.

Table 7—Heat from People (at 80 F)

1. Average condition (at rest, eating, office work, etc.)		
Sensible heat	220	Btu per hour per person
Latent heat	180	" " " " "
Total	400	" " " " "
2. Medium rate of exertion (restaurant waiters, average dancing, etc.)		
Sensible heat	230	Btu per hour per person
Latent heat	420	" " " " "
Total	650	" " " " "

Determination of Total Heat Gain

Selecting the Time of Day for Calculation

In selecting the time of day for calculations, one must distinguish between two general types of application. The first class includes all applications which do not have an unusually large and sharply variable heat gain from people or appliances, such as residences, offices, some types of retail shops, etc. The second class would include all applications which do have a large heat gain from people and appliances, varying sharply at certain hours of the day, such as restaurants, night clubs, theatres, funeral parlors, etc.

Cases in the first category may be handled quite simply and with reasonable accuracy. The time of maximum heat gain in such cases is dependent upon the climatic factors and degree of exposure of the space being air conditioned. It does not depend so much on variables like occupancy and heat producing appliances. Because of this it has been possible to make a study of the probable time of maximum heat gain in such cases with various exposures, and reach substantially accurate conclusions beforehand. The results of this study are given in Fig. 6 and Table 8. It is necessary merely to select from Fig. 6 that room which has the same exposure as the room being calculated, and then refer to Table 8 to find the correct time of day for that room.

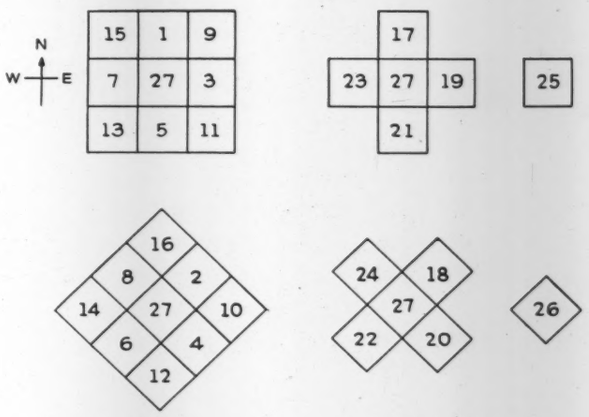


Fig. 6—Typical room exposures (Use with Table 8 to determine time of maximum cooling load)

Table 8—Time of Maximum Cooling Load for Rooms with Different Exposures

(To be used with Fig. 6)

Room Number from Fig. 8	Number of Walls Exposed	Exposed Walls	Awnings on Windows: Occupied Space Above	Awnings on Windows: Roof or Attic Above	No Awnings on Windows: Occupied Space Above	No Awnings on Windows: Roof or Attic Above
1	1	N			2 p.m.	2 p.m.
2		NE			2	2
3		E	2 p.m.	2 p.m.	9 a.m.	9 a.m.
4		SE	1	1	10	10
5		S	2	1	1 p.m.	1 p.m.
6		SW	3	2	4	3
7		W	3	3	4	4
8		NW	4	3	5	4
9	2	N E	2	2	9 a.m.	9 a.m.
10		NE SE	2	1	9	9
11		E S	2	1	10	10
12		SE SW	3	2	3 p.m.	3 p.m.
13		S W	3	2	3	3
14		SW NW	3	3	4	4
15		W N	3	3	4	4
16		NW NE	4	3	5	5
17	3	W N E	4	3	4	4
18		NW NE SE	3	3	4	4
19		N E S	2	2	10 a.m.	10 a.m.
20		NE SE SW	3	2	3 p.m.	3 p.m.
21		E S W	3	2	3	3
22		SE SW NW	3	3	4	4
23		S W N	3	3	4	4
24		SW NW NE	4	3	4	4
25	4	S W N E	3	2	3	3
26		SW NW NE SE	3	2	4	4
27	None				2	2

It is not intended that Fig. 6 and Table 8 should be used blindly. This standard method was worked up on the basis that the rooms under consideration were approximately square; that outside walls were exposed to the sun, and not shaded by other buildings; that outside walls had one or more windows; that the heat from occupants and appliances was small as compared with the heat from other sources. Consequently, this data does not apply exactly to rooms which do not conform to these conditions, although it is quite correct for rooms which do not deviate too widely from these specifications.

As an example of the use of Fig. 6 and Table 8, let us consider a room which has a south and west exposure, no awnings on the windows, and an occupied space above, such as might occur in a modern office building. The sun effect on the south wall and windows reaches a maximum at noon. Sun effect on the west wall and windows reaches a maximum at 4 p.m., and the value thereof is greater than for the south wall. The

outdoor temperature, and consequently the tendency to gain heat by conduction through practically all walls, ceilings, floor, etc., is greatest about 3 p.m. The heat from outdoor air also reaches a maximum about 3 p.m. From this it is apparent that the effects at 3 p.m. or thereabouts, will overshadow the effects at noon. Furthermore the sun effect on the west wall at 3 p.m. will be almost as great as at 4 p.m. Consequently 3 p.m. should be the proper time for which to make the calculation. It will be noted that room 13 in Fig. 6 fulfills the specifications laid down above, and from Table 8 the time of maximum heat gain is listed as 3 p.m.

Now suppose that this same room were located under a roof or a hot attic. The sun effect above the room would be a large factor and would reach a maximum at noon, just the same as on the south wall. The effect of this would be to make the maximum total heat gain come somewhat earlier than before—probably about 2 p.m. instead of 3 p.m. This is shown in Table 8.

Let us suppose now that there is under consideration a restaurant doing its greatest business at dinner, and therefore having its maximum occupancy between the hours of 6 and 8 p.m. The people would naturally supply a considerable proportion of the total heat gain and would probably control the time of day when the total heat gain is maximum. Hence, it is not correct to make the calculations for 2 or 3 p.m., and then add the heat gain from occupants which occurs at a later hour. On the other hand, it would not be quite accurate to make calculations of all components for 6 or 7 p.m., because of a factor which has been mentioned previously, but not discussed, namely, the effect of time lag or retardation of the heat flow through walls. This factor will be discussed in more detail later. The correct solution in the case of a restaurant of this nature would be to calculate the heat gain from sun effect through windows, lights and appliances, ventilation air and occupants at 6 or 7 p.m. (of course, in the particular case being discussed there would be no sun effect through windows at this late hour), and to add to them the conduction through walls and sun effect through walls calculated for about 4 p.m. or earlier.

In case of doubt as to the proper time to use, the following procedure is recommended:

1. Calculate the maximum value which each component attains during the day, assuming that there is no time lag in the heat transmission through walls.
2. Note the hours at which these peaks occur.
3. Consideration should be given to time lag phenomena in walls, in a manner to be discussed later.
4. Select the probable time of maximum total heat gain from an analysis of the above data.
5. Where the economics of the application warrants the additional time required, an hour by hour calculation of the heat gain may be made.

The Effect of Heat Storage in the Walls

Very little has been said thus far about the subject of heat storage in walls, yet it would seem that it must certainly have some effect on the heat gain of a structure. What, then, is the justification for ignoring it in the method of calculation which has been described? The answer is that this effect has not been ignored, but its discussion has been deferred in order to not complicate the picture unduly. Actually, in a great many cases the effect of heat storage in walls may be ignored without creating a serious error in the calculations. Whether this effect may or may not be ignored depends largely on the relationship between the magnitudes of the conduction and sun effect through walls and the other components of the heat gain. If the two former components are predominant, then it may be desirable to take account of the effect of heat storage. On the other hand, if these two components are small as compared with the rest of the heat gain, the neglect of heat storage phenomena will produce no serious error.

The method of calculating conduction and sun effect through walls, described, has been compared with some very accurate tests[10] conducted a few years ago on walls having various heat capacities, and has been shown to give highly accurate results for walls which have heat capacities comparable to those found in ordinary building construction. For walls having abnormally high capacity the calculations give results which are slightly on the conservative side. This comparison demonstrated that the heat capacity of normal walls has relatively little effect on the amount of heat transmitted through them, but it does give rise to a time lag of one hour or more, depending on the construction.

Until more complete experimental evidence is available on the difference of time existing between the manifestation and maximum outdoor effects and maximum heat gain from walls of various building materials, and on the amount of reduction in total heat conducted on account of the storage effect, the following procedure is recommended:

1. Calculate the heat from sun effect through windows, appliances, lights, ventilation air and people on the basis of temperatures, sun effect factors, and other conditions actually existing at the time of maximum total heat gain.
2. Add to the above conduction through walls and sun effect through walls calculated on the basis of temperatures existing from 3 to 6 or more hours earlier in the day than the time at which the maximum total heat gain occurs. Reference may also be made to Chapter 8 of THE A.S.H.V.E. GUIDE[3] for time lag factors.

Unusual Sources of Heat Gain

The obvious sources of heat which have been discussed above are sometimes supplemented by hidden and unexpected sources, the possibilities of which must not be overlooked. One fairly obvious one is the conduction of heat through the walls of the ducts in a central plant system, particularly where the ducts run through unventilated attics, outside walls, etc. The amount of heat thus conducted to the conditioned air can be calculated from a knowledge of the temperatures inside and outside the duct, and from a consideration of the usual heat transfer phenomena. In general, the following formula may be used with sufficient accuracy.

$$H_d = A_d\ U_d\ (t_1 - t_2) \tag{12}$$

$$U_d = \frac{1}{\frac{1}{1.2} + \frac{t}{k}} \tag{13}$$

[10]Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by Houghten, Blackshaw, Pugh and McDermott, A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 231.

where

H_d = heat transmitted through walls of duct system, Btu per hour
A_d = area of duct surface, sq ft
t_1 = temperature of air outside the duct, F
t_2 = temperature of air inside the duct, F
t = thickness of insulation applied to duct, inches.
k = conductivity of duct insulation, Btu per hour per sq ft per F per inch thickness.

Another unusual source of heat gain arises when rooms or furred spaces are used as plenum or return air chambers. Sometimes the walls of such chambers adjoin very hot spaces, like kitchens or attics, and sometimes they are exposed to the outside. When such cases are encountered, it is necessary to calculate the heat gain of the air chamber as well as the heat gain of the air conditioned space.

A third source of unusual heat gain arises when steam pipes are hidden in the walls and carry steam during the summer time. This might happen in hospitals or various industrial plants, for example. Such cases as this are rare but have been known to result in the maintaining of unsatisfactory indoor conditions when their existence was not known before the equipment was installed. Calculation of heat gain from such sources may be made on the basis of the temperatures involved and the usual heat transfer phenomena.

Methods for Reducing Heat Gain

The application engineer should always be on the alert for opportunities to reduce the heat gain of a structure where means are available to do so in such a manner as to reduce the net cost of the installation over the period of its life.

The use of insulating materials in walls which have a heavy heat gain is one obvious method of reducing the size of equipment required. The use of awnings, blinds or shades is an aid in the reduction of sun effect through windows. Of course, such shades, blinds, etc. are of no benefit unless they are drawn. The spraying of water on roofs which are directly over the air conditioned spaces is another method which has been tried successfully at times. This water may sometimes be taken from the water discharge circuits of the condensing unit or from an independent circuit. Attic ventilation is another means of effectively reducing the heat gain through the roof. In many cases, unventilated attics situated beneath roofs which are exposed to the full intensity of the sun may rise to temperatures as high as 130 or 140 F. Since the outdoor air is seldom hotter than 110 deg, it has the capacity to remove heat from such attics, provided it is forced through by a suitable fan or blower. The fan should be able to provide in the neighborhood of 30 complete air changes per hour in the attic space. Under such conditions the average air temperature in the attic space would usually be from 5 to 10 deg warmer than outdoors during the middle of the afternoon, when the sun effect is greatest. There are times, of course, when attic insulation may be more valuable than attic ventilation.

Summary and Conclusions

This paper has described a complete and systematic method for determining the character and magnitude of the cooling load for air conditioning systems. The salient features of this method may be summarized as:

1. It organizes and systematizes the work so that a complete and accurate calculation requires a minimum of both time and technical experience.
2. A new and simple method is presented for accurately calculating the sun effect through opaque walls and roofs.
3. The method of separating the components of the total load is such that the total sensible and total latent heat gains may be determined for any hour of the day.
4. An approximate method of allowing for time lag effects resulting from heat storage further improves the accuracy of the calculations.
5. A procedure is given for predetermining the time of day when the total heat gain is greatest.
6. The general procedure is applicable to an hour-by-hour calculation of the cooling load, if the economics of the application warrants it.
7. Although the discussion has been centered around the determination of the maximum total heat gain, it is equally adaptable to the determination for various conditions of operation, and may be used to determine the maximum ratio of dehumidification to sensible cooling.

The method of making heat gain calculations described herein has been under development for several years, during which time it has undergone many refinements of detail. However, it has been used in fundamentally the same form as presented here for over 2 years by a large number of application and sales engineers. The experience gained from the many installations made on the basis of cooling loads calculated in this manner has substantiated its simplicity and reliability.

The method is not perfect. There is a large field for improvements. Some of the points which need elaboration and refinement are:

1. The effects of heat storage in walls on the cooling load when a constant indoor temperature is maintained.
2. The effects of heat storage on the pull-down load after the cooling system has been shut down for some time, or following a sharp change in outdoor temperature.
3. More complete data on infiltration rates during the cooling season.
4. More complete data on the load reducing properties of various types of shading devices, expressed as a function of the type of material and age.
5. More complete and accurate data on the ability of various outside building surfaces to absorb solar radiation, expressed in terms of the type and age of the surface, and the angle of the surface with respect to the rays of the sun.
6. A more complete analysis of climatic data.

It is hoped that further research will provide the information necessary for making additional refinements in the cooling load calculations.

Bureau of Standards Sound Tests

The equipment and methods used in making sound transmission measurements at the National Bureau of Standards are described in a paper published in the June issue of the Bureau's *Journal of Research*.

Specifications for the construction of the test floor and wall panels, on which measurements were made, are included.

The results of sound transmission measurements on a number of floor and wall panels are given. The paper also makes known results on the transmission of impact noises for floor panels and briefly describes methods of reducing this type of noise.

An All Electric Heating, Cooling and Air Conditioning System

Installation and Performance of Plant Using Entirely the Heat Pump Principle

By **Philip Sporn*** ***(NON-MEMBER)*** **and D. W. McLenegan**** ***(MEMBER)***

Introduction

THE use of a refrigerating machine as a heat pump for the heating of a house in winter is not new, having been first suggested more than 80 years ago by Lord Kelvin[1]. Others [2], [3], have pointed out that at least in southern climates refrigerating machines installed for the purpose of cooling houses in the summer, might, by the use of the above principle, also be installed for the heating of houses in the winter-time. In recent years a number of heat pump installations have been built and described [4], [5], and at least one installation made embodying the idea of using a single set of refrigeration equipment to carry out the heating and cooling with the same equipment[6].

As is well known, the ideal Carnot efficiency of a reverse refrigeration cycle operating as a heat pump is given by the equation $E = \frac{T_1}{T_1 - T_2}$. Under this arrangement a heat pump taking heat from a source at a temperature of 60 F, and discharging heat at a temperature of 135 F, would have an ideal efficiency of 7.95. With the outside temperature at —20 deg, the ideal efficiency drops to 3.74. With the actual efficiency running in the order of 60 per cent of the ideal, it is apparent that under climatic conditions where temperatures of the order of zero to —20 deg have to be encountered, the efficiency of a heat pump drops very rapidly and may approach a value close to unity. The net effect of this is one of two things:

(a) If the heat pump is selected with a view of being adequate under the extreme low temperatures, then a size of compressor is involved that is so much larger than that necessary for refrigeration, that it is not only uneconomical from the standpoint of equipment investment cost, but is also out of balance from the standpoint of having approximately the same equipment performing the job in summer and in the winter; or

(b) If an attempt is made to avoid that condition, then some provision has to be made for taking care of the extreme cold weather conditions by auxiliary means. These may take the form of fuel heaters or direct electric heaters, and some of the installations[6] have resorted to these methods.

In adopting that solution, however, there is definitely raised the question of whether the heat pump as a method of heating is a self-reliant piece of equipment or could even be developed as such. The particular installations previously described obviously made no attempt to answer these questions or to solve the problem of economical balance between heating and cooling equipment. In brief, none of the heat pump installations previously made have been complete, or all heat pump installations.

It has been apparent for some time to those responsible for the development described in this paper, that if electric heating of homes, stores, offices and similar buildings is to overcome the handicap of, first, a 70 per cent heat loss in going from thermal to electrical energy, involved even in the best steam generating plant, and, second, even more important, the excessive costs of large distribution facilities necessary in any straight 1:1 heat conversion process when going from electrical to thermal energy; and if electric heating is to overcome the handicap of the fixed charges involved in cost of equipment, then the reverse refrigeration cycle of heating, which makes possible the use of the same equipment for cooling in summer and heating in the winter, and which gives the maximum economy in use of electrical energy, with a minimum requirement of electrical capacity to take care of a given installation, would be able to overcome all such handicaps. Hence, in considering the heating, cooling and air conditioning system for the particular building described in this paper, the first decision made was that it would be an all electric and all compressor system. Having decided that point, and in order to meet the difficulties with regard to unbalance between summer and winter refrigerator capacity already pointed out, and the difficulties of the concomitant heavy investment cost that

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[1] Thomson, William (Lord Kelvin), On the Economy of Heating and Cooling of Buildings by Means of Currents of Air, *Glasgow Phil. Soc. Proc.*, v. 3, December, 1852.

[2] Stevenson, A. R. Jr., Cooling Homes, A Field for Refrigeration, presented at the symposium of the Refrigeration with Gas Committee of the *American Gas Association*, April 20, 1926.

[3] Stevenson, A. R. Jr., Refrigeration, *Journal of the Franklin Institute*, Vol. 208. No. 2, Aug., 1929, p. 143 and following.

[4] Haldane, T. G. N., The Heat Pump—An Economical Method of Producing Low-grade Heat from Electricity, *Electric Review*, v. 105, pp. 1161-1162, December 27, 1929, and *I. E. E. Journal*, v. 68, pp. 666-675, June, 1930.

[5] Wilkes, Gilbert and Marbury, R. E., House Heated by Pump with 5 to 1 Pick-up Ratio, *Electrical World*, Vol. 100, December 17, 1932, p. 828 and following.

[6] Doolittle, H. L., Edison Building Heated and Cooled by Electricity, *Power*, v. 74, pp. 348-351, September 8, 1931.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Toronto, Ont., Can., June, 1935.

[6] Doolittle, H. L., Edison Building Heated and Cooled by Electricity, *Power*, v. 74, pp. 348-351, September 8, 1931.

would be involved in an all compressor system if the source of heat were the outside atmosphere (since the building located in southern New Jersey had cycles of zero F and even lower temperatures to contend with) two possible solutions were considered:

(1) A storage system. With this system heat obtained by the heat pump from the outside air under favorable (mild temperature) conditions would be stored at a moderate temperature (between 60 deg and 80 F) using that portion of the compressor capacity not required for immediate heating. This storage would be made available as a source of heat for the compressors under conditions of exceedingly low outside temperatures when conditions for obtaining of heat directly by the heat pump from the outside were unfavorable, and

(2) A more simple system, in which the heat would be obtained from water supplied at a constant temperature from a deep-well.

Both of these solutions were thoroughly explored with the definite knowledge that at least in northern climates the first of them would be more general and of greater applicability than the second. However, after considering the various technical problems involved which had theretofore not been definitely worked out, it was decided not to attempt too much in the first installation and to try the simpler of the two solutions, that is, utilize a deep-well water supply as the source of heat in the winter and as a sump for the rejection of heat in the summer-time.

The Building

The building heated, cooled and air conditioned by the equipment described in this paper is a two-story and basement, modern but very simple, brick and steel reinforced concrete structure belonging to the Atlantic City Electric Co., a subsidiary of the American Gas and Electric Co., and located at Salem, New Jersey. The lower floor is devoted to a display room, service department and offices, while the entire second floor is used as a combined demonstration hall and auditorium. The two photographs reproduced as Figs. 1 and 2 show clearly the character of the building. The latter, too, shows very distinctly the arrangement of grilles at the base of the walls for exhausting the cold air and the registers along the ceiling for delivering the conditioned air to the room. The basement of the building is used for storage and contains the pump used to give the water supply, and the air conditioning equipment. Fig. 3 shows the well and pump installation.

The brick walls and roof are insulated with 2 in. of cork. There are 1,240 sq ft of windows. The total volume of the building is approximately 76,800 cu ft, of which the first and second floor represent a volume of 55,200 cu ft.

From the building plans, the heating requirement for 0 F outdoor temperature was found to be:

	Btu/Hour
Conduction loss, first and second floors	143,000
Warming the ventilation air for first and second floors	107,000
Basement heat loss	35,000
Total	285,000

The heat loss of the basement does not vary greatly with outdoor temperature, being affected principally by the ground temperature, which is constant.

For summer operation, it was planned to cool the first and second floors to 80 deg dry-bulb temperature with 50 per cent relative humidity, assuming outdoor conditions at 95 deg dry-bulb and 75 deg wet-bulb. Allowing for 1200 cfm of ventilation air, the maximum summer load is 108,000 Btu/hour distributed as follows:

	Btu/Hour	
Sensible Heat		
Conduction	33,300	
Solar Radiation	18,300	
Ventilation air	19,600	
40 people	8,800	80,000
Latent Heat		
Ventilation air	20,800	
40 people	7,200	28,000
Total		108,000

The total air flow was based on the following conditions:

(1) Air flow to be adequate for maximum winter heating at relatively low discharge temperature.

(2) Summer cooling to be accomplished with air not more than 15 deg below room temperature.

Fig. 1—Salem office building

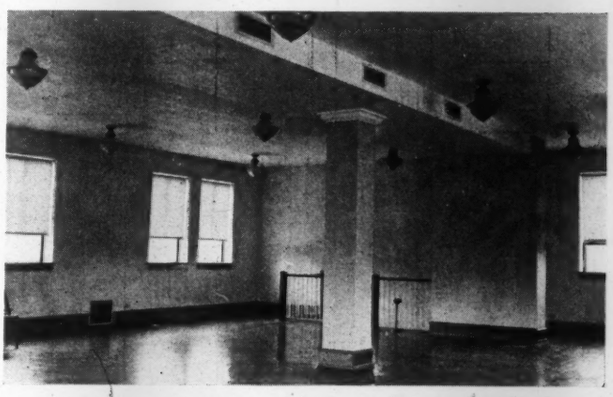


Fig. 2—Interior view showing inlet and outlet grilles

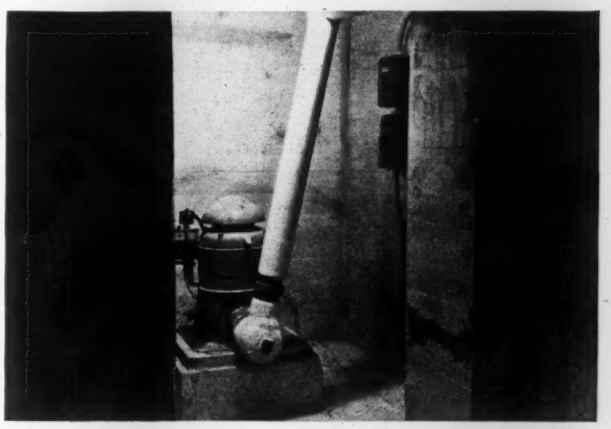


Fig. 3—Well and water pump

(3) Air flow to be small enough to avoid excessive duct sizes.

A total air flow of 5500 cfm was selected for conditioning the first and second stories of the building, which represent a volume of 55,200 cu ft. The basement is heated by the machine *losses*, but is not air conditioned. Thus, air is circulated through the first and second floor

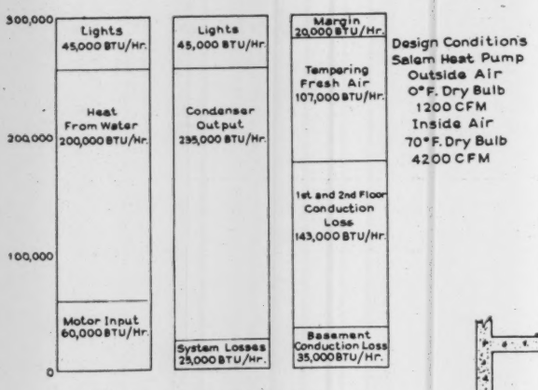


Fig. 4—(above) Heat balance diagram

Fig. 5—Schematic layout of heating system

space at the rate of one change every 10 min, consistent with liberal practice. The system is designed for the admission of 1200 cfm of outdoor air, with provision for increasing this quantity to 1700 as desired. The average occupancy of the building is about 25 people. The ventilation air intake thus represents 48 cfm per person for summer or winter conditions, which is rather high. However, in terms of the maximum expected, which is 60, it represents a value of only 20 cfm per person and it was felt that until actual experience indicated that less ventilation air could be employed, it was not desirable to go below this value.

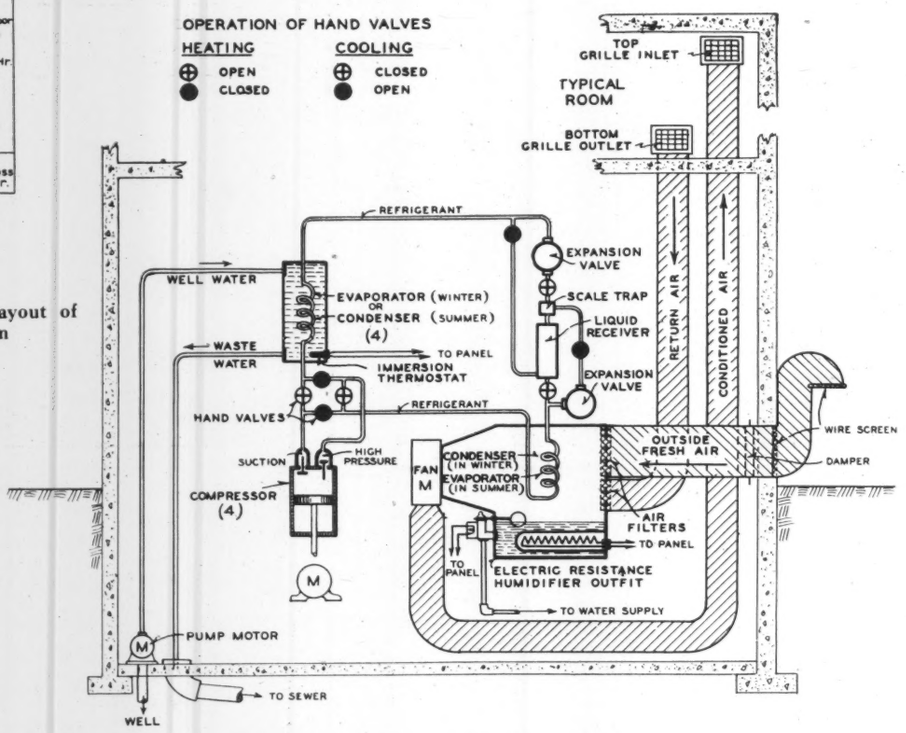
With 0 F outdoor temperature, the heat balance was planned as in Fig. 4. This diagram shows that the heat pump equipment alone is not quite adequate for a sustained period of zero weather, since the capacity was planned to be 260,000 Btu/hour and the maximum requirement was 285,000. It was felt, however, that the selection of this capacity was justified, since the building has a large lighting load which could be called on in extreme cases. Furthermore, the building is well insulated and would not require the maximum heating effect, except during a prolonged period of zero weather, which very seldom occurs.

The System

The system comprising the air conditioning installation is shown schematically in Fig. 5 and a description of its operation is given.

A deep-well pump with 3 hp, 3-phase motor, forces water through 4 water coolers in series. As refrigerant evaporators, these units are connected to 4 compressors and thence to the condensers (air heaters) as separate refrigerant circuits. Units Nos. 2 and 3 are arranged with reversible connections for summer operation, and for simplicity, these 2 machines are always operated as a single unit, since they are adequate to carry the summer cooling load.

Outdoor air entering the system passes through a preheating coil, not shown in the diagram of Fig. 5, which is the condenser of heat pump No. 1. This unit, controlled by a thermostat in the entering air, operates continuously when the outdoor temperature is below



40 deg. The ventilation air is then mixed with the recirculated air, and passes in succession over the condensers of Nos. 2 and 3 and No. 4 heat pumps, absorbing heat to a degree consistent with the heating requirements. An electrically heated humidifier adds moisture to the air, under control of a humidistat. The air is then distributed through the building through a conventional high velocity duct system, using overhead horizontal delivery and baseboard returns. The velocity in the supply mains is 1400 fpm, in the branches, approximately 1000, and at the outlets approximately 750, which permits adequate travel and mixing with the room air.

Since all air delivery, whether heated or cooled, enters through high wall registers, the air distribution had to be planned to avoid stratification during the heating season. This problem was successfully solved by locating a number of ample return openings at floor level, to remove the cooler air, and by using a relatively large total flow of air in proportion to the heat delivered. Under the most severe conditions, the temperature of the air delivered does not exceed 105 deg; and the air at this temperature is not too light to mix readily. With continuous circulation and a minimum cyclic variation of the air delivery temperature, it was

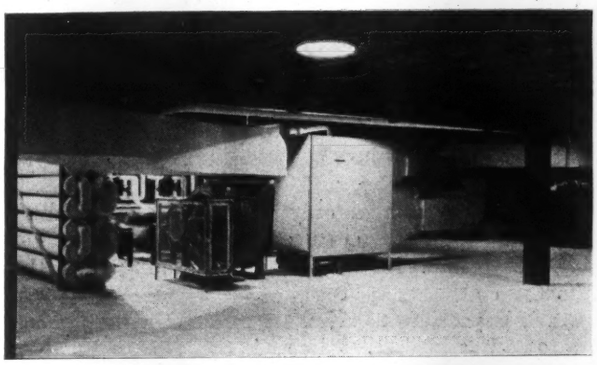


Fig. 6—Closed view of condenser and humidifier (heating cycle)

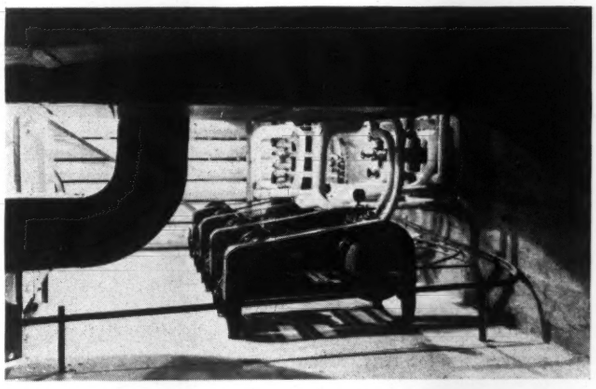


Fig. 7—Compressor installation

expected that comfortable conditions would be maintained, as regards both the vertical temperature variation and the temperature-time variation. Under maximum load conditions, the refrigerant condensing temperature of the last heating unit (No. 4) was planned not to exceed 135 F.

Considering the heat pump only, greater efficiency could have been achieved by delivering air at a lower maximum temperature, since the compressors could then have been operated at lower condensing temperatures and hence with lower power input. This, however, would have required larger ducts, since a greater air flow would have been needed for the same heat delivery. Conversely, the use of a smaller quantity of hotter air would have impaired the heat pump performance, as well as the vertical temperature distribution. The authors believe that the outlet temperature selected represents a good compromise for this installation, although climate, building construction and further design and operating experience may call for a modification of this figure.

The compressors are 4 cylinder, single acting units, operating at 400 rpm, each belted to a 5 hp, 3-phase, 60 cycle, 1800 rpm induction motor, and are of a type commonly used in comfort cooling installations. Although the air-cooled condensers, which heat the conditioned air in winter, are not built integrally with the compressors, each compressor has its own liquid receiver, through which the refrigerant passes before returning to the evaporator.

The sizes of the air-cooled condensers have been proportioned to give efficient operation, taking into account also the first cost of this surface. The condenser for machine No. 1, which operates only as an air preheater in winter, consists of 4 coil sections and is relatively small, since the air entering through this condenser is cold and the unit operates at low discharge pressure. Machines Nos. 2 and 3 each have a 5-section condenser which the air enters at an average temperature near 60 deg in winter. Condenser No. 4 consists of 10-sections, since it must dissipate heat to air which has already been somewhat heated in cold weather by units Nos. 2 and 3; hence, this surface is made large to reduce the refrigerant-to-air temperature difference, and thereby improve the coefficient of performance of this machine.

A general view of the physical arrangement and appearance of the equipment involved is shown in Fig. 6. A view of the 4 compressors is shown in Fig. 7. Fig. 8 shows a detailed view of the evaporators (water

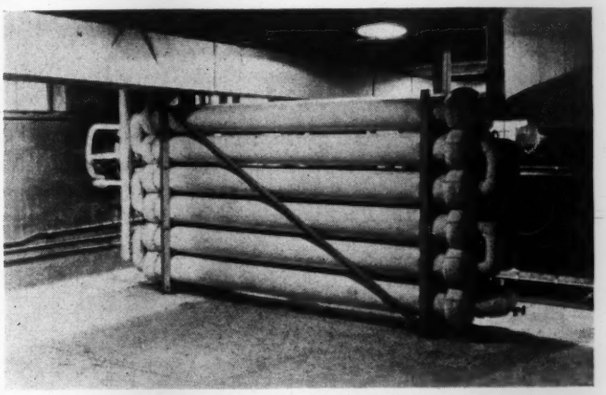


Fig. 8—Evaporators (heating cycle)

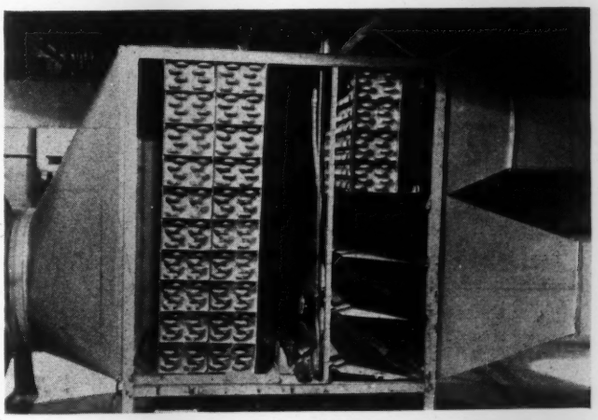


Fig. 9—Open view of condenser and humidifier (heating cycle)

coolers) and Fig. 9 shows the condensers (air heaters) including the filtering and humidifying arrangement.

The operating sequence of machines to meet the heating demand is shown in Fig. 10. An indoor thermostat starts and stops the machines, but their availability is determined by separate thermostats, responsive to outdoor temperature and located in the incoming air stream. Thus, machine No. 4 runs alone and intermittently at outdoor temperatures above 54 deg. From 54 deg to 40 deg, machines 2 and 3 are also available. Below 40 deg machine No. 1 is added and runs continuously. Below 23 deg machines Nos. 1 and 4 operate continuously and Nos. 2 and 3 are available.

The purpose of the sequencing scheme is to operate at all times with a minimum of excess capacity, and hence with lowest possible refrigerant condensing temperature. This has two advantages, namely:

(1) For any required average rate of heat output, continuous operation with low condenser pressure yields a higher coefficient of performance than intermittent operation at high condenser pressure and higher momentary output.

(2) It is desirable from the comfort standpoint to have continuous air delivery, using slightly heated air in mild weather, and increasing the air temperature in colder weather. In contrast, a single machine, or 4 machines always operated concurrently, would cause the hottest air to be delivered intermittently in the mildest weather.

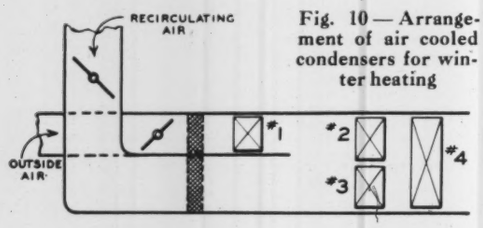


Fig. 10 — Arrangement of air cooled condensers for winter heating

The remainder of the control scheme is shown in Fig. 11 and centers about this selection of units. The air conditioner fan is operated continuously and at constant speed. By a system of interlocks, the water pump is started automatically whenever any of the compressors are started by the indoor thermostat. In case of pump failure, the water coolers are protected from freezing by immersion thermostats (with 40 deg setting), located in the lower sections of the water cooler. The humidifier in the air conditioner is started and stopped directly in response to a room humidistat.

The changeover from winter to summer operation involves:

(a) Transfer of the control circuit to reverse the operation of the room thermostat. For summer operation, a cooling rather than a heating requirement is indicated when the thermostat closes its contacts.

(b) Manipulation of valves. The air-cooled condensers now become refrigerant evaporators, in which the flow is regulated by automatic expansion valves. These valves, normally by-passed in winter, are cut in by switching to a different refrigerant manifold connection. The water coolers now become water-cooled condensers, hence the expansion valves of these units are by-passed. By reversing the flow through the liquid receiver, the process of reversing the refrigerant cycle is completed.

As indicated by the schematic diagram Fig. 5, this change is easily accomplished. Distinguishing colors have been used for the winter and summer valves, to aid the operator.

Tests

During the test period in February, 1935, the equipment was operated as a heating plant. The outdoor temperature varied between 33 F and 46 F, hence the equipment was operating between one-third and one-half of its maximum capacity, and the tests cover the case of full capacity operation only for short periods during which the indoor temperature was allowed to rise slightly above normal. However, by determining from these tests the constants of the various units of the equipment, it is possible to calculate accurately the performance at other load conditions. This is true particularly because the temperatures of the water-heated evaporators are confined to a narrow range, and do not depend directly on the temperature of the outdoor air.

Measurements taken in the tests were:

Water

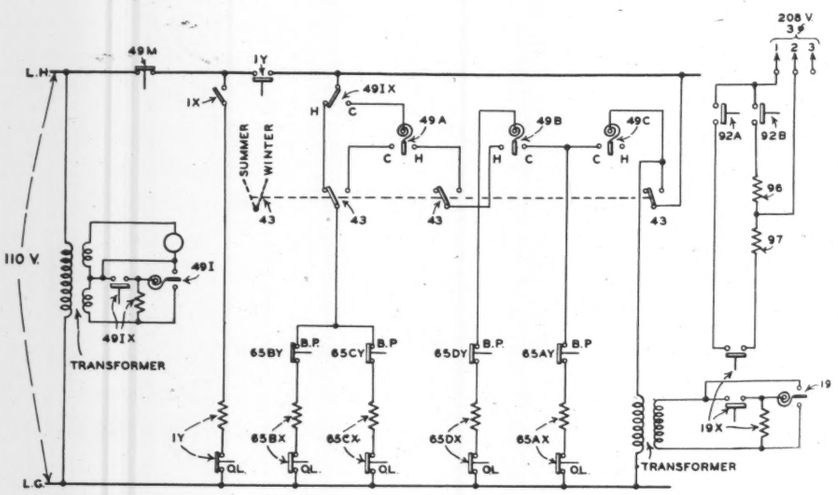
(a) Rate of flow.

(b) Temperature of water entering and leaving each cooler. These measurements were made by thermo-couples soldered to the water pipes; also by thermometers located in wells, and these readings were checked by bleeding off water at three points (inlet, intermediate and outlet) using both thermo-couple and thermometer measurements.

These readings represent one method of measuring heat absorbed by the evaporators.

Refrigerant

(a) For each of the 4 heat pump units, suction and discharge pressures were read, and converted to equivalent temperatures. Readings of temperature were taken at each evaporator outlet and at each compressor intake and discharge to indicate the degrees of superheat at these points. Temperatures at each condenser inlet, and outlet were recorded, and also liquid temperatures before the expansion valves, to indicate the degree of sub-cooling of the liquid refrigerant. These points determine the refrigerant cycle, and are used in conjunction with standard compressor test data, to determine the heat output of the machines. Incidentally a



INDEX TO SYMBOLS

1X - FAN CONTROL SWITCH	65AX - COMPRESSOR CONTROL CONTACTORS
1Y - " " CONTACTOR	65BX - " " "
19 - HUMIDISTAT	65CX - " " "
19X - HUMIDISTAT RELAY	65DX - " " "
43 - TRANSFER SWITCH	65AY - BACK-PRESSURE CONTACTS ON COMPRESSOR
49A - OUTSIDE THERMOSTAT	65BY - " " " " "
49B - " "	65CY - " " " " "
49C - " "	65DY - " " " " "
49I - INSIDE "	92A - FLOAT SWITCH
49IX - " " RELAY	92B - " "
49M - IMMERSION THERMOSTAT CONTACTS	96 - VALVE SOLENOID
	97 - HEATER COIL

Fig. 11—Schematic diagram of heating system

further check based on compressor speeds, displacements and clearances agreed closely.

Air

(a) Total air flow through the heating coils (refrigerant condensers) was obtained by Pitot tube traverses in the conditioner discharge duct.

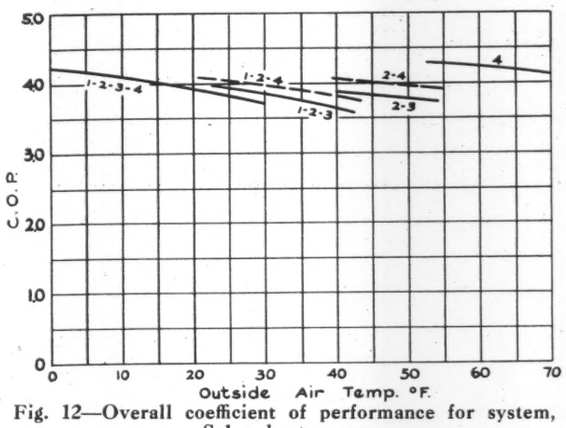


Fig. 12—Overall coefficient of performance for system, Salem heat pump

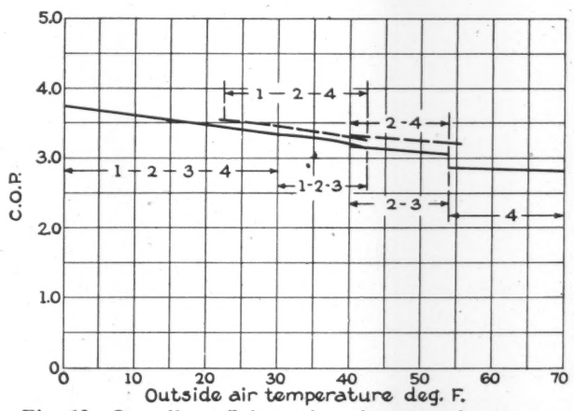


Fig. 13—Overall coefficient of performance for system, including water pump input, Salem heat pump

(b) The intake of outside air was measured by traverses with an electric hot-wire anemometer, the velocities being too low to permit accurate measurement with a Pitot tube.

(c) Temperature of outside air entering was measured by three thermo-couples and a recorder. An aspirating psychrometer, reading both wet-bulb and dry-bulb temperatures, gave a measure of relative humidity of entering air.

(d) The temperature of air leaving the preheater was recorded, to provide a measure of preheater output.

(e) The condition of recirculated air was measured by means of thermocouples and an aspirating psychrometer.
The amount of recirculated air can be obtained from the measurements of total flow and outside air intake.

(f) Temperature of air leaving the conditioner was measured to provide a check on the performance of units Nos. 2, 3, and 4.

Electrical

(a) Kw input to each compressor motor.

(b) Kw input to fan motor.

(c) Kw input to pump motor.

(d) Kw input to electric humidifier.

(e) Total kw and kwhr during each test.
Voltage remained at the normal value of 208 during the test.

Outputs of the units as determined from the air measurements agreed closely with those determined from the refrigeration cycle. Capacities determined from water flow measurements checked within 10 per cent, which is reasonably good in view of the small differences in water temperature. However, the water measurements were used only as a rough check on the other methods.

In determining the performance of the system, the power used by the air circulating fan and by the electric humidifier have not been taken into account. Fan and humidifier elements are part of any winter air conditioning system, and are not directly related to the method of generating heat for the building. Using the equipment for heating, rather than cooling, the coefficient of performance has been calculated from the test data on two bases:

(1) (Condenser output plus machine losses) divided by (Electrical input to the compressors.)

(2) (Condenser output plus machine losses) divided by (Electrical input to compressors *and* water pump.)

The work of compression is included as useful output. The machine losses also may justly be included in the useful heat output since this heat is used to maintain temperature in the basement, which is used only for files and storage, and is not air conditioned.

Method (2) is really the correct criterion of performance, since the pump is an essential part of this particular system. The pump power increases the total power input, but does not increase the heating effect within the building, except by the small amount of the motor losses. Hence, the coefficient of performance is lowered by including this item.

Fig. 12 indicates that, without considering the pump, the coefficient of performance for the composite equipment averages about 3.9, and varies somewhat with outdoor temperature, *i.e.*, with the number of machines in operation. This chart also shows that the performance is best at the extremes of the operating range. In mild weather, only one of the 4 units operates, and the unit having the largest condenser is logically selected for this service, since it can operate with the lowest refrigerant condensing temperature. This and the high evaporator temperature of a single machine with maximum water flow yield a high coefficient of performance. As units are added for colder weather, more heat is extracted from the water, and some of the units must therefore operate at lower evaporator temperatures with poorer performance at intermediate outdoor temperatures. However, as lower incoming air temperatures are encountered, the preheater and the units immediately following are able to operate at very low condensing temperatures and the high coefficient of performance of these individual machines raises the average of performance of the group in very cold weather. With air entering at 0 F, the preheater unit alone develops a coefficient of performance of 5.0.

As indicated by the dotted lines of Fig. 12, the order in which machines are added affects the performance by as much as 10 per cent at certain outdoor temperatures. As would be expected, best performance is obtained by using as a base load machine the one having the largest condenser, and by adding machines in such order that the largest remaining condenser surface is associated with the lowest available temperature of entering air.

Fig. 13 shows the coefficient of performance including water pump power. The coefficient of performance is not greatly reduced at heavy loads (low outdoor temperatures) since the water pump input is only a small

percentage of the total electrical input. At light load (one compressor operating) the coefficient of performance is materially reduced by this fixed item of water pump power. However, the energy represented by this light load operating condition is only a small percentage of the total energy used to heat the building throughout the winter.

Performance

The performance throughout the entire heating season of 1934-35 has been as expected, the equipment functioning perfectly with few minor exceptions enumerated. The lower floor of the building, being the office of an electric utility in a fairly substantial community, presents a rather difficult problem in maintaining constant levels in heating and air conditioning. With people walking in and out almost continuously, and with a wide variation in traffic over a weekly or monthly interval, it would be expected that a particularly heavy strain would be placed on the regulating and air conditioning equipment. Nevertheless, throughout the entire period an equable and proper temperature was maintained un-

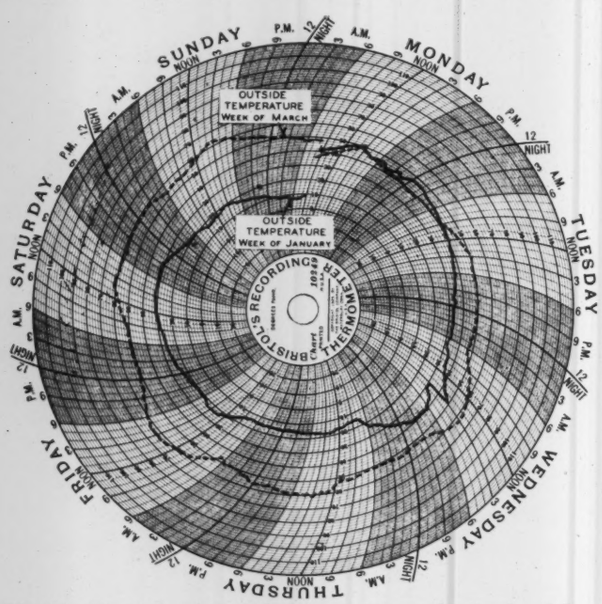


Fig. 14—Chart showing outdoor temperatures for weeks of Jan. 21, 1935—Jan. 28, 1935 and Mar. 18, 1935—Mar. 25, 1935

der all conditions of outside temperature. Figs. 14 and 15 illustrate this point very clearly, Fig. 14 representing a week of rather severe weather conditions for the climate during the week January 21-28, 1935, and another week of much milder and more even temperature during the week of March 18-25, 1935. The corresponding indoor temperatures are shown in Fig. 15. As will be noted, in spite of the wide divergence in outside temperatures and some quite low temperatures for the territory, the inside temperatures were maintained on almost even scales for corresponding periods in the two weeks with no overloading in any way on the equipment.

Although integrated results of the entire winter's operation are not yet available, monthly data up to the end of April have been obtained. The data given in Table 1 covering the month of February, are indicative of a typical month representing neither the extreme low temperature nor the extreme mild weather conditions encountered.

Table 1—Actual Performance Data On Salem Heat Pump January 31 to February 28, 1935

Outdoor Temperature	Deg
High	55.
Low	10.
Average	34.5
Indoor Temperature	
High	78.
Low	56.
Average	70.
Incoming Water	Average 57.
Outgoing Water	" 53.
Water Consumption	915,300 Gal
Electric Consumption	Kwh
Fan Motor, 2 hp	582
Pump Motor, 3 hp	1502
Humidifier	1503
Compressor, No. 1	2085
" " 2	1370
" " 3	1283
" " 4	1015
Total Metered	9475

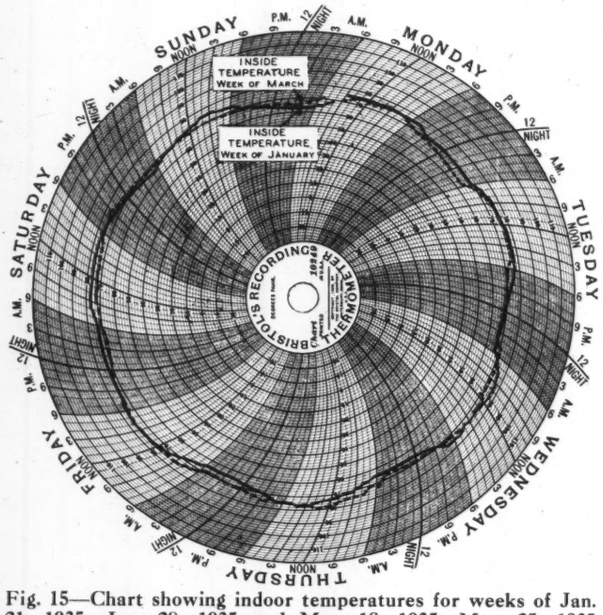


Fig. 15—Chart showing indoor temperatures for weeks of Jan. 21, 1935—Jan. 28, 1935 and Mar. 18, 1935—Mar. 25, 1935

Based upon the data presented as to heat losses in the building, the coefficient of performance is calculated on the basis of both debiting the heat pump with the pump power and on the basis of neglecting this, and gives coefficients of performance of 3.51 and 4.45, respectively. From the data in Fig. 13, it would be expected that the performance under an average condition of 35 deg would be approximately 3.3 as compared with the measured value of 3.51, and the performance actually obtained is sufficiently close to that, the higher performance being probably due to the effect of solar radiation not taken into account in the calculation of heat losses.

It is expected that the performance at the end of this season will actually be somewhat better than obtained during the last season, due to the fact that the tests have indicated that slight variation in the unit combination employed over the temperature range will give somewhat better performance. This is shown graphically in Fig. 16, which indicates the heating capacity (condenser output of the units) in comparison with the load. The lower load curve represents the heat loss of the air conditioned space. The upper curve includes the heat loss of the basement. With all units operated, the heat given off by the motors, drives and compressors, amounts to about 32,000 Btu/hour, which is very close to the heating requirements of the basement. Fig. 16 also indicates that there is some over-capacity at all times, but the sequence of units reduces this capacity to a relatively

small amount. By using a larger number of units, it is obvious that the capacity of the compressors could be made to conform still more closely to the load, but it was not believed that the additional cost and complication would be justified. However, it will be possible by changing the combinations indicated in solid lines to those indicated in dotted lines, to improve the efficiency

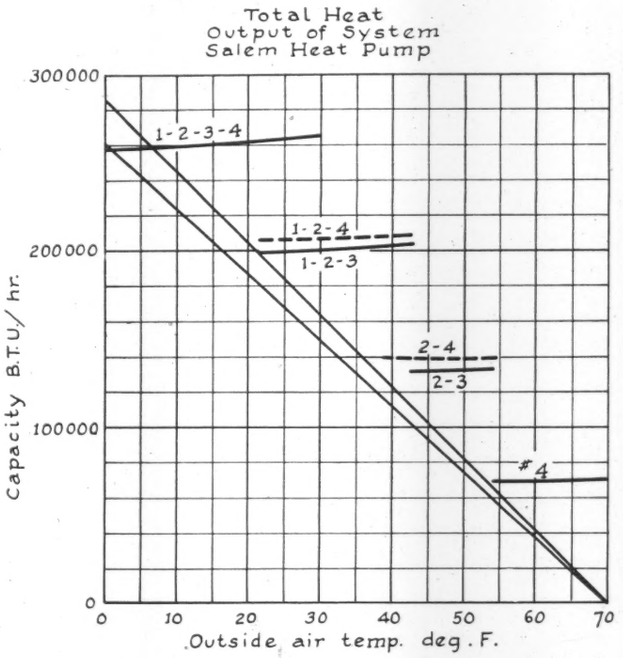


Fig. 16—Total heat output of system, Salem heat pump

and thus bring up the annual coefficient of performance and it is expected to make this change for the next season's heating requirements.

A few minor difficulties were experienced during the winter season. These consisted of ordinary adjustment troubles on compressors, gas joints, belts and pulleys, but all of these were easily corrected. Perhaps the most serious difficulty encountered was the tripping of the water pump motor due to an incorrect operation of the thermal overload relay, resulting in a stoppage of water under a condition where the compressors were running. This resulted in frozen evaporators and it was necessary to take the compressors out of service for a day to thaw the evaporators out. This, however, has been taken care of by a change in the control wiring, giving the proper interlock to prevent a repetition of this difficulty.

In summer, using only machines Nos. 2 and 3, the system will operate as a conventional direct expansion system with water cooled condensers. These 2 units meet the cooling requirements and will be operated intermittently under room thermostat control with continuous air flow. Neglecting the water pump input, a coefficient of performance of 3.0 is expected; including the water pump, coefficient of performance will be approximately 2.3. These figures are based on the cooling effect produced, rather than on condenser heat delivery. The kw demand of the system in the summer, including the pump, will be approximately 55 per cent of the winter demand, although the maximum summer load in Btu is only 40 per cent of the maximum heating load.

Due to the many new elements involved which had to be analyzed for this particular installation, it is difficult to evaluate correctly the initial cost of this type of system as compared with more conventional plants. One item, the cost of heat transfer surfaces, is inherently high since the condenser temperature must be low compared to steam coil temperatures to permit efficient operation of the heat pump. However, with the experience gained from this installation, it is expected that the cost of future installations of this type should not greatly exceed that of equivalent year round systems, which use combustion equipment for heating and mechanical refrigeration for cooling. For installations in milder climates where the maximum cooling and heating loads are more nearly equal, the excess cost of the reverse cycle equipment needed for heating tends to disappear completely.

The Future

While it was fully realized that the particular working out of the heat pump principle described in this paper would have rather limited direct application, nevertheless it is a fact that there are many cases in this country where such applications can be directly made. In fact, the surprising fact to the authors has been the number of cases that have developed where duplicate applications can be made and, apparently, to advantage. A number of these installations in fact are being figured on at the present time and there is every reason to believe that those which will be installed will be fully as successful, both from a technical and economic standpoint, as the present installation. Experience both in test and performance of the installation described herein dictates no substantial, and only a limited number of minor, changes for future applications of the same type.

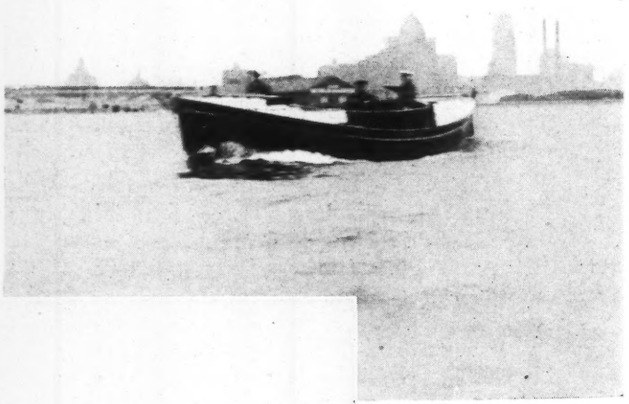
The development obviously has excellent possibilities for extensive application. Whether the extent is so large as to render reasonable the looking forward to its application on an economical basis in a million homes is questionable. As pointed out previously in this paper, such an eventuality is unlikely of realization with the present form of this development; but the steps from the form in which this case was carried out to the more general case, where storage would be involved, are, while not particularly simple, nevertheless definitely attainable: the means for carrying out these steps have been definitely indicated from the tests here described. It is the intention, therefore, just as soon as an installation suited for full air conditioning, and representing within itself all the problems likely to be encountered in a general application of that kind, is available, to tackle and work out the complete solution. The authors hope and expect that such an installation will actually be made within the next year and hope further to be able to present the results of such a general solution before the membership of this Society.

The authors wish to acknowledge the cooperation and help received from members of the staffs of the two companies with which they are affiliated in carrying out this installation and in making the tests described and, more particularly, to record their thanks to Mr. A. M. Newman of the Atlantic City Electric Co. for his help in carrying out the tests and in following up the operation of the equipment, and Mr. H. Gibson of the General Electric Co. for his help in working up the test data.

Group of A. S. H. V. E. Members Embarking for Toronto Harbor Tour

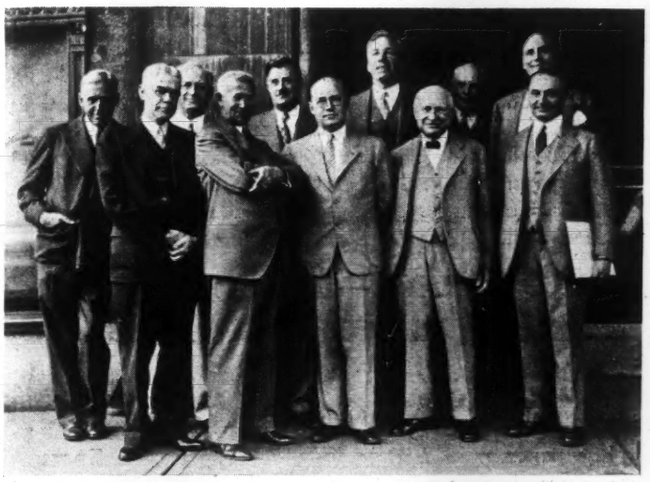
Second Launch as seen from First Launch

High Speed Life-Saving Launch of Toronto Police Dept.



Life Saving Surf Boat

Harbor Board Tug Rouille Demonstrating Fire Fighting Equipment



Officers and Members of the Council
(*Left to right*): E. H. Gurney, R. H. Carpenter, G. L. Larson, C. V. Haynes, J. F. McIntire, W. E. Stark, John Howatt, J. D. Cassell, M. C. Beman, F. C. McIntosh and A. J. Offner

Proceedings of the Semi-Annual Meeting 1935

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

Toronto, Ont., Can.

THE members of the Society responded with enthusiasm to the invitation of Ontario Chapter and a fine three-day meeting was enjoyed at the Royal York Hotel, Toronto, June 17-19. Members came from the Atlantic to the Pacific and many made this trip the occasion for a vacation tour, visiting Ottawa, Montreal, Quebec, Thousand Islands, Sault Ste. Marie, and other points of historic interest.

On each of the three days in Toronto many sessions were held to discuss technical papers and reports. Attendance at the meetings was exceptionally heavy, indicating that the Program Committee had selected subjects that were of unusual interest.

FIRST SESSION—

Monday, June 17, 1935, 10:00 A. M.

Pres. John Howatt called the 1935 Semi-Annual Meeting of the Society to order in the Ballroom of the Royal York Hotel, Toronto, and M. W. Shears, President of the Ontario Chapter, made a brief speech of greeting.

President Howatt expressed the appreciation of the members for the welcome given on behalf of the Ontario members.

The first item of business was consideration of two amendments to the By-Laws which were prepared by the Committee on Constitution and By-Laws and approved by the Council and submitted to all members, in accordance with Article B-XVI as follows:

Add the following new section to Article B-VIII:

Section 11. At the first meeting of the Council after the Annual Meeting, or as soon thereafter as possible, the president shall appoint a Committee of at least five (5) members, whose duty it shall be to arrange for the publication of The Guide. This Committee shall be responsible for the text section and shall arrange for the inclusion of such new chapters as will furnish information regarding new developments.

In order that the wording of the various Sections of Article B-VIII may be in agreement, the Council suggested a change in the wording of the present *Section* 11, which would become *Section* 12.

Section 11. At the organization meeting of the Council, each year, the president shall appoint a Committee of five (5) members to be known as the F. Paul Anderson Award Committee. This Committee shall function according to the terms of the F. Paul Anderson Award. (See Appendix B.)

To Be Amended as Follows: Section 12. At the first meeting of the Council after the Annual Meeting or as soon thereafter as possible, the president shall appoint a Committee of five (5) members to be known as the F. Paul Anderson Award Committee. This Committee shall function according to the terms of the F. Paul Anderson Award. (See Appendix B.)

It was moved and seconded that the new *Section* 11 of Article B-VIII and the amended *Section* 12 be adopted. This motion was unanimously carried, and President Howatt announced that these new sections would become effective immediately.

The first technical paper, Characteristics of Registers and Grilles, by J. H. Van Alsburg, was presented by the author. (Complete paper published in June 1935, JOURNAL SECTION, *Heating, Piping and Air Conditioning*). A brief discussion of this subject was offered by John Everetts, Jr., New York.

President Howatt announced that the Research Laboratory of the Society had made several studies in school rooms and the results were presented in three papers by Director F. C. Houghten. With the aid of lantern slides Mr. Houghten presented the following papers: Classroom Odors with Reduced Outside Air Supply, by F. C. Houghten, H. H. Trimble, Carl Gutberlet and M. F. Lichtenfels; Air Supply to Classrooms in Relation to Vent Flue Openings, by F. C. Houghten, Carl Gutberlet and M. F. Lichtenfels; Classroom Drafts in Relation to Entering Air Stream Temperature, by F. C. Houghten, H. H. Trimble, Carl Gutberlet and M. F. Lichtenfels. (Complete papers published in May, June and July 1935, respectively, JOURNAL SECTION, *Heating, Piping and Air Conditioning*). A written discussion on the subject of classroom odors by Dr. C.-E. A. Winslow, Director of

the John B. Pierce Laboratory of Hygiene, New Haven, Conn., was read by John James.

A rather lengthy discussion of this subject was given by Messrs. W. L. Fleisher, New York, W. H. Driscoll, New York, A. J. Nesbitt, Philadelphia, J. H. Walker, Detroit and J. H. Van Alsburg, Holland, Mich. At the conclusion of the discussion Mr. Houghten answered the various questions which had been asked.

President Howatt had a special word of greeting for three of the Society's Life Members who were in attendance, J. D. Cassell, Philadelphia, J. M. Stannard, Chicago, and Warren Webster, Camden, N. J. Just before adjournment, announcement of the voting on the Garage Ventilation Code was given. A tabulation made by the Board of Tellers, Messrs. H. W. Fiedler, W. M. Heebner and H. B. Eells gave the following results: *for* the Code 215, *against* the Code 5, *total legal votes* 220, *total votes cast* 222.

SECOND SESSION—

Tuesday, June 18, 1935, 9:30 A. M.

The Second Session of the Semi-Annual Meeting was called to order by President Howatt on Tuesday morning and the first technical paper, entitled Heat Absorbing Glass Windows, by W. W. Shaver, was given by the author. (Complete paper will be published in a later issue of *Heating, Piping and Air Conditioning.*) Those who discussed the subject were W. W. Timmis, New York, and S. R. Lewis, Chicago, and a reply was made by Mr. Shaver.

C. P. Creighton, Montreal, co-author of the paper, Cooling and Air Conditioning in an Ale Brewery, by C. P. Creighton and F. J. Friedman, was introduced to the members and made the presentation. (Complete paper published in July 1935, JOURNAL SECTION, *Heating, Piping and Air Conditioning.*)

A description of the operation of an unusual heating and cooling system was given by D. W. McLenegan, co-author of a paper, entitled An All Electric Heating, Cooling and Air Conditioning System, by Philip Sporn and D. W. McLenegan. (Complete paper published in August 1935, JOURNAL SECTION, *Heating, Piping and Air Conditioning*). Comments were made by R. S. Libby, Vancouver, B. C., and H. E. Turner, New York.

The concluding paper on this morning's program was given by F. O. Urban, one of the authors who prepared A Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads, by F. H. Faust, F. O. Urban and L. Levine. (Complete paper published in August 1935, JOURNAL SECTION, *Heating, Piping and Air Conditioning*). Verbal discussions of this paper were given by W. H. Carrier, Newark, N. J., W. W. Shaver, Corning, N. Y., and Prof. M. K. Fahnestock, Urbana, Ill.

THIRD SESSION—

Wednesday, June 19, 1935, 9:30 A. M.

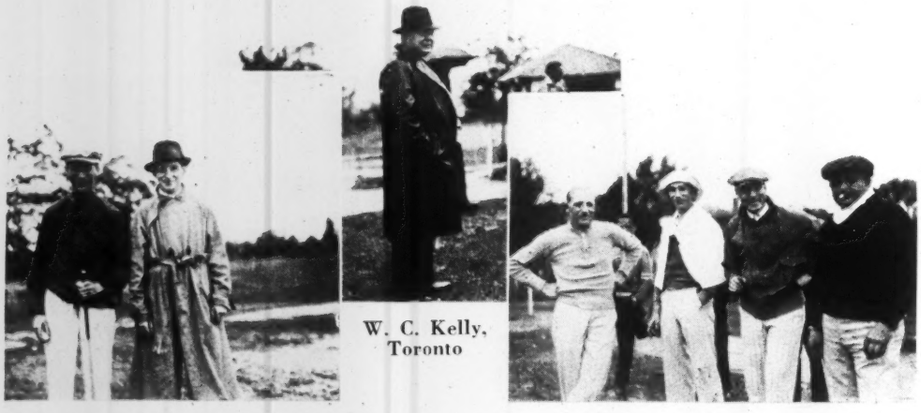
When the Wednesday morning session was opened, Pres. John Howatt called on Prof. D. W. Nelson, who presented the paper, Oil Burning in Residences. (Complete paper published in July 1935, JOURNAL SECTION, *Heating, Piping and Air Conditioning*). Lively discussion followed this presentation and those who participated were S. R. Lewis, Chicago, H. M. Hart, Chicago, Prof. F. B. Rowley, Minneapolis, M. B. Watson, Toronto, Col. W. A. Danielson, Washington, D. C., and R. S. Libby, Vancouver.

The second paper, The Dust Problem in Air Conditioning, by Prof. F. B. Rowley, was presented by the author. (Complete paper published in June 1935, JOURNAL SECTION, *Heating Piping and Air Conditioning*). A written discussion by H. C. Murphy, Louisville, commended the paper and said that Professor Rowley had made a valuable contribution, pointing out the difficulty in correlating results under different test procedures. Additional comments were given by H. M. Hart, Chicago.

S. R. Lewis read his paper, Chicago Standards for Certified Air Conditioning, which commented on proposals that a special committee had suggested as standards for that city. (Complete paper will be published in a later issue of *Heating, Piping and Air Conditioning.*) Extensive discussion of the subject was participated in by C. A. Dunham, Chicago, S. H. Downs, Kalamazoo, Prof. D. W. Nelson, Madison, Wis., J. H. Van Alsburg, Holland, Mich., H. M. Hart, Chicago, and Col. W. A. Danielson, Washington, D. C.

Proposed Amendments

Under the heading of new business, F. C. McIntosh submitted two amendments to the Constitution, relating to Student Membership and explained that the lower age



W. C. Kelly, Toronto

J. J. Everetts, Jr. and W. L. Fleisher, Jr., of New York

G. E. Otis, Moline, Ill., W. M. Hughes, Toronto, A. J. Nesbitt and H. M. Goldberg, Philadelphia

limit suggested would be helpful to those technical schools and colleges which were extending their courses in heating, ventilating and air conditioning. The following amendments were presented on petition of 19 members.

ART. C-II, *Section 7.* A Student Member shall be a person between the ages of 20 and 25 years, who is regularly attending courses in an engineering college or technical school at the time of applying for membership.

To Be Amended as Follows: A Student Member shall be a person between the ages of 18 and 25 years, who is regularly attending courses in an engineering college or technical school at the time of applying for membership.

ART. C-III, *Section 4.* When a Student Member discontinues his regular studies in an engineering college or technical school, it shall be incumbent upon him to apply within one (1) year for advancement to either Junior or Associate grade. In no case shall the period of Student Membership exceed four (4) years.

To Be Amended as Follows: When a Student Member discontinues his regular studies in an engineering college or technical school he shall be automatically transferred to Junior grade of membership unless he has applied for and has been elected to Associate grade. In no case shall the period of Student Membership exceed four (4) years.

On motion, properly seconded, it was

VOTED: TO refer this matter to the Committee on Constitution and By-Laws for presentation to all members of this Society in the prescribed manner.

Tribute to Stewart A. Jellett

C. V. Haynes, Chairman of a special committee appointed by President Howatt, offered a resolution on the late Stewart A. Jellett who was a Charter Member, an Honorary Member and the first man to serve a full term as President of the Society.

WHEREAS, Stewart A. Jellett was a Charter Member and the first man to serve a full term as President of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and

WHEREAS, the Society has honored him by conferring Honorary, Life and Presidential Membership upon him as a recognized leader in the development and advancement of heating and ventilating, and

WHEREAS, he was a pioneer in the profession and proved his ability as an engineer and executive, and

WHEREAS, he also served the Society as secretary on committees and in an advisory capacity during its 41 years of existence, and

WHEREAS, it has been the will of Almighty God to take him unto Himself, therefore

BE IT RESOLVED, that we, the members present at the Semi-Annual Meeting 1935, in Toronto, Canada, hereby express our deepest regret at his passing but rejoice in having been associated with one of nature's noblemen who endeared himself to all of our members, and

BE IT FURTHER RESOLVED, that this resolution be inscribed upon the proceedings of the Society and a copy be presented to the bereaved family so that they may know with what high esteem his memory is held.

C. V. HAYNES, *Chairman.*
H. P. GANT,
J. F. HALE.

On motion, properly seconded, this resolution was unanimously adopted.

Resolutions

O. J. Prentice, Chicago, presented formal resolutions of thanks and appreciation for the splendid handling of the meeting and the enjoyable program that had been provided.

The Committee on Resolutions has a particularly happy privilege. This privilege, however, even at technical sessions, presents a problem. The Committee feels that suitable appreciation of the wise planning, the untiring energy and the self-sacrificing devotion of those who have made this meeting so outstanding a success cannot be expressed in a formal—"Be it resolved,—whereas, and whereas, and whereas" but, that such expression should emulate the friendliness which has characterized these stimulating three days.

BE IT RESOLVED, that the thanks and appreciation of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in Convention assembled be expressed—

1. *To the Ontario Chapter,* for a hospitality superlative alike in its graciousness, in its completeness and in its never failing anticipation of our every comfort and need. A hospitality that has been the sum total of the individual contribution of each member of the Chapter and which finds its completeness in the never to be forgotten friendly and efficient contribution by the ladies.

2. *To the City of Toronto,* which through its officials and home loving citizens has made us feel at home and which, through the whole hearted cooperation of the Police Department in transportation matters and in the unfailing courtesy of the policemen on the streets, has made our sojourn a happy experience in a city of brotherly love.

3. *To the Convention Bureau and the Local Press,* for the publicity given to our Society and to the Sessions of this Semi-Annual Meeting in particular.

4. *To the Ontario Research Foundation, the Harbor Commission and the Various Clubs and Organizations,* which have so generously added to the enjoyment of our visit.

5. *To the Authors of Papers,* for their investment of time in preparation and for their presentation of important subjects touching the welfare of humanity.

6. *To the Trade Press,* for its advance publicity of this Meeting, and for its constant interest in the work of the Society and publication of its doings.

7. *To the Royal York Hotel and its staff,* for a practical interpretation of the friendliness of the city, and of the Ontario Chapter, an interpretation which has made our residence here most comfortable.

O. J. PRENTICE, *Chairman.*
R. V. SAWHILL,
W. E. STARK.

On motion, properly seconded, these resolutions were unanimously adopted. The meeting adjourned.

Council and Committee Activities

The Council held two meetings in Toronto, the first a dinner meeting on Sunday, June 16, at 6:30 p. m., with the following members present: John Howatt, President, M. C. Beman, R. H. Carpenter, J. D. Cassell, E. H. Gurney, C. V. Haynes, G. L. Larson, J. F. McIntire, F. C. McIntosh, A. J. Offner, W. E. Stark and A. V. Hutchinson, Secretary.

Special reports from Officers and Committees on the current condition of Society activities were received. The Finance Committee was instructed to change reserve fund investments to U. S. Government obligations, in accordance with the Constitution and By-Laws.

The desirability of a joint session with *A. S. R. E.* was discussed and it was suggested that the two Societies meet in June, 1936, at a resort to be mutually agreed upon.

The Council recommended to all Chapters that their Constitution and By-Laws be revised so that Chapter Officers will be elected and installed at the last meeting in the spring to permit time for planning meeting programs during the summer season.

A report on the sale of exhibit space for the 4th International Heating, Ventilating and Air Conditioning

Top: H. J. Church and R. G. Heard of Toronto, W. H. Baker, Chicago, and T. K. Dobie, London
Center: C. V. Haynes, Philadelphia, R. V. Sawhill and R. E. Moore, of Chicago
Bottom: H. H. Erickson, J. H. Hucker and A. E. Kriebel of Philadelphia, and R. C. Bolsinger, Norristown, Pa., winner of the Research Cup.

Exposition, to be held in Chicago, Ill., January 27 to 31, 1936, was presented to the Council.

In accordance with the By-Laws, the Council set the schedule of dues for 1936 and waived initiation fee for all grades of membership. Members' and Associates' dues for 1936 will be $18 annually, Junior Members' $10, and Student Members' $5 if they receive THE GUIDE, otherwise, the annual fee for Students will be $3.

Five candidates were nominated to serve on the Committee on Research for a three-year term commencing 1936: Col. W. A. Danielson, Washington, D. C., C. E. Lewis, Dayton, O., Prof. D. W. Nelson, Madison, Wis., Cyril Tasker, Toronto, and Dr. C.-E. A. Winslow, New Haven, Conn.

During the Semi-Annual Meeting two meetings of the Committee on Research were held and several decisions on the question of policy were considered and the present status of Research problems was discussed by those in attendance, who included Prof. A. P. Kratz, *Chairman*, Urbana, Ill., C. A. Booth, Buffalo, S. H. Downs, Kalamazoo, C. A. Dunham, Chicago, W. L. Fleisher, New York, H. C. Murphy, Louisville, J. H. Walker, Detroit, F. C. Houghten, Director, Pittsburgh, Pres. John Howatt, Chicago, M. K. Fahnestock, Urbana, Ill., E. H. Gurney, Toronto, S. R. Lewis, Chicago, D. W. McLenegan, Schenectady, E. D. Milener, New York, and J. H. Van Alsburg, Holland, Mich.

A luncheon meeting of the Committee on Ventilation Standards was held on Tuesday, June 18, with the following members present: W. H. Driscoll, *Chairman*, New York, J. J. Aeberly, Chicago, W. H. Carrier, Newark, N. J., S. R. Lewis, Chicago, H. M. Hart, Chicago, and by invitation, John Everetts, Jr., New York.

At 1:30 p. m. on Tuesday, Prof. G. L. Larson, *Chairman*, Guide Publication Committee, called a meeting to order with the following members present: S. H. Downs, Kalamazoo, W. L. Fleisher, New York, E. N. McDonnell, Chicago, W. W. Timmis, New York, J. H. Walker, Detroit, Pres. John Howatt, Chicago, and John James.

On Tuesday at noon, a get-together meeting of chapter officers and representatives was held and for several hours problems of mutual interest on local chapter administration were considered. Pres. John Howatt called the meeting to order and the following representatives discussed subjects of interest: Messrs. J. H. Hucker, and H. H. Erickson, Philadelphia, W. E. Voisinet, Buffalo, S. H. Downs, Kalamazoo, Ernest Szekely and C. H. Randolph, Milwaukee, R. J. Tenkonohy, St. Louis, J. H. Milliken and J. J. Hayes, Chicago, A. C. Wallich, Detroit, and T. F. Rockwell, Pittsburgh.

A general discussion followed and the meeting voted to recommend that future conferences of similar character be held each year.

On Wednesday at noon, a luncheon meeting of the Nominating Committee was held and its report will be given in the October issue of the JOURNAL.

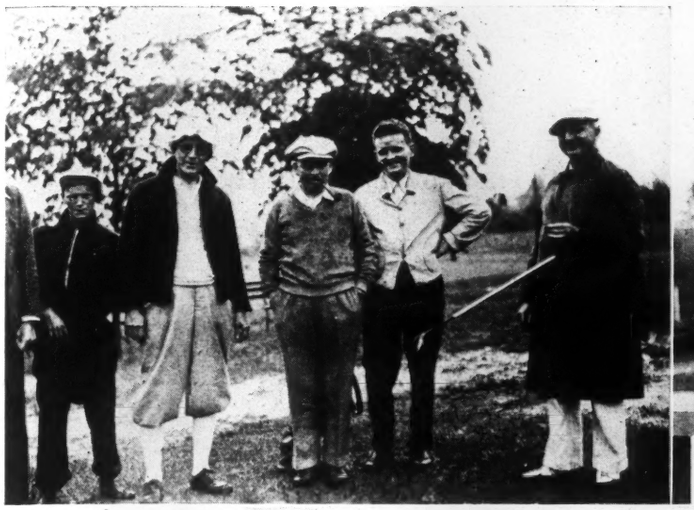
Entertainment

With an attendance of close to 350, the Society's Semi-Annual Meeting, 1935, at the Royal York Hotel, Toronto, was an outstanding success and tribute to the efforts of the Ontario Chapter members who were hosts for the occasion.

E. H. Gurney, member of the Council, was General Chairman of the Committee on Arrangements and was assisted by L. L. Anthes as Vice Chairman, and all plans for the meeting were carried out under the direction of the various sub-committee chairmen who were responsible for the numerous entertainment events.

H. H. Angus and his Reception Committee were ready to greet the arriving members and ladies early on Monday morning, June 17, and J. S. Paterson and his Committee handled the details of registration.

At one o'clock even though the weather was not favorable, 40 golfers left the Royal York Hotel for the Weston Golf Club to play in a tournament for the Research Cup. Between and during the showers, the golfers battled Old Man Par and when the firing ceased, R. C. Bolsinger of Norristown, Pa., was eligible to carry home the Research Cup. The tournament was under the direction of W. P. Boddington, Chairman of the Entertainment Committee, and he was assisted by Messrs. H. J. Church and L. D. McKellar.



Above: M. W. Best, Hamilton, D. J. Macdonald, Toronto, E. K. Campbell, Jr., Kansas City, and N. D. Adams, Rochester, Minn.

Below: G. C. Crawford, Toronto, C. H. Turland, Winnipeg, and D. C. Simpson, Newark, O.

Above: J. P. Kiely, Chicago, S. W. Alexander, Toronto, and E. W. Rietz, Chicago

Below: E. N. McDonnell, Chicago, M. F. Blankin, Philadelphia, H. M. Hart, Chicago, and R. F. Connell, Detroit

Golf trophies were awarded for low net and low gross scores in three classifications A, B, and C, according to handicap as well as several special awards which were made as follows: W. E. Voisinet, Buffalo, J. P. Kiely, Chicago, R. G. Heard, Toronto, D. C. Simpson, Newark, O., E. N. McDonnell, Chicago, H. J. Church, Toronto, W. L. Fleisher, Jr., New York, G. C. Crawford, Toronto, M. W. Best, Hamilton, R. E. Moore, Chicago, and E. K. Campbell, Jr., Kansas City, and A. E. Kriebel, Philadelphia, were tied for the most strokes.

At three o'clock on Monday afternoon M. W. Shears, Chairman of the Transportation Committee, had private cars ready to take the ladies for a sight-seeing trip through the city of Toronto and a visit to the Granite Club. At 6:30 p. m. cars were again ready to take members and ladies to the Old Mill, where they joined the golfers for an informal dinner and dance. After a delightful steak dinner, everyone enjoyed dancing until a late hour.

On Tuesday morning, Mrs. E. H. Gurney and Hugh Jenney, who shared the responsibility of chaperoning the ladies, organized a group who visited the Ontario Museum to inspect the Chinese porcelain exhibit. In the afternoon the ladies enjoyed a sail across the bay to the Royal Canadian Yacht Club where many played bridge, after which tea was served. Two inspection trips had been arranged for the men—one at the Ontario Research Foundation, and the other in the form of a boat trip to the Toronto harbor development. During this trip the fire boats gave a demonstration and a life saving drill was staged for the members.

The big social event of the meeting was the Semi-Annual Banquet and Dance in the ballroom of the Royal York, with 300 present. The guests at the head table marched in behind a Scotch piper and the toastmaster, E. H. Gurney, said Grace, after which there were the customary toasts to the King and the President of the United States. Those at the head table were: Pres. and Mrs. John Howatt, Mr. and Mrs. E. H. Gurney, Prof. and Mrs. G. L. Larson, B. A. Trestrail, A J. Offner, J. D. Cassell, J. M. Stannard and Prof. A. P. Kratz. At the conclusion of an excellent dinner, the toastmaster intro-

duced the Past Presidents of the Society and the members of the Committee on Arrangements: L. L. Anthes, *Vice-Chairman,* H. H. Angus, *Reception,* Mrs. E. H. Gurney and H. B. Jenney, *Ladies,* W. P. Boddington, *Entertainment,* Thomas McDonald, *Banquet,* E. R. Gauley, *Publicity,* M. F. Thomas, *Finance,* J. S. Paterson, *Registration,* M. W. Shears, *Transportation,* and W. R. Blackhall, Past President, and H. R. Roth, Secretary, of the Ontario Chapter. The speaker of the evening was B. A. Trestrail who told in his inimitable way how some of the important world problems could very quickly be solved. The audience particularly enjoyed his solution for the world depression and his plan for bringing prosperity to Canadian railways. After the ballroom had been cleared for dancing the orchestra played popular numbers. Between dance numbers, two Scotch pipers played and special Scotch dances were given by a group of four talented dancers. Credit for the success of this event goes to Tom McDonald and his hard working committee.

Wednesday morning some of the ladies went on a shopping tour of the Toronto department stores and many souvenirs were purchased.

When it came time to end the Semi-Annual Meeting 1935, those who had enjoyed the hospitality of the Ontario Chapter members were enthusiastic in their praise of the splendid work of the Committee and the enjoyment that they had derived during their three-day stay in Toronto.

N. P. Fenner Heads Heating Department

Announcement has recently been made of the appointment of N. P. Fenner as head of the heating department recently established by John G. Kelly Co., 210 East 45th St., New York City. Mr. Fenner is well qualified for his new duties through his association with Hoffman Specialty Co., Inc., during the past 16 years.

The John G. Kelly Co., who have been the Metropolitan New York distributor for Hoffman Specialty Co., Inc. venting valves, took over the sale of the balance of the company's specialties and pumps on July 1.

Cleveland Rotary Club Elects Walter Klie

The Rotary Club of Cleveland, Ohio, chose a member of the heating industry as their leader for the coming year in the election of Walter Klie. Mr. Klie is a well known heating contractor and he has been president of the Smith & Oby Co., Cleveland, O., since 1913.

Mr. Klie has been active in society and association activities and has served a three-year term as a member of the Committee on Research of the A. S. H. V. E. He is also a past president of the *Heating, Piping & Air Conditioning Contractors National Association* and of their local Cleveland association, and of the *National Association of Building Trades Employers.*

Wisconsin Chapter Picnic

The Wisconsin Chapter held its annual picnic on Thursday afternoon, June 27, at The Chalet, about 15 miles north of Milwaukee on the shore of Lake Michigan and a good time was had by all. There was an attendance of 37 at dinner, and your correspondent saw a good many of the old members who seldom attend chapter meetings, but do attend picnics.

The Committee in charge of the picnic, John S. Jung Chairman; A. M. Wagner and F. G. Weimer, had a fine sports program arranged, and a good many who have attended every picnic concede that this one was the best yet. One of the features was that Chairman Jung worked out a plan whereby the picnic was not only self-sustaining but produced a small surplus for the chapter treasury, which was very gratifying to the officers.

The annual ball game, which is looked upon as more important than the Big League All Star game, was the first item on the program. The Captains of the rival teams were Victor Berghoefer and Edgar Gerstenberger. After a furious nine

PRESIDENT'S MESSAGE TO MEMBERS

SOLAR RADIATION

The month of August is one of burning sun and scorching heat, a month in which we are made conscious of the effects of solar radiation and the advantages of controlling its absorption by structures occupied by humans.

Studies of the flow of heat through the walls of an enclosed space have engaged the interest of our Society and its Research Committee since their organization. The rapid development in space cooling has served to focus attention on this problem in recent years.

Solar radiation and its effect on heat loss from, and heat gain to, buildings has proven a complicated problem because of the different characteristics of the materials used in building construction. A great mass of facts, however, is gradually being built up that will eventually provide data that will permit predetermination of the effects of solar radiation within reasonable accuracy.

Our Research Laboratory and the universities cooperating with our Society have conducted studies on the intensity of energy transmission through solar radiation, its variation during hours of sunshine, the magnitude of the effect on different exposures of a building, the rate of transfer of heat of solar radiation through different types of structures, the rate of absorption of solar radiation by surfaces of different color and characteristics, the rate of heat transfer by solar radiation through windows and the effects of curtains, shades, awnings, etc. These studies will have their application in determining the effects of solar radiation on different building exposures in reducing the heating load in winter and in increasing the cooling load in summer.

It is difficult to change the habits and customs of a people. We have been in the habit of taking solar radiation as one of nature's phenomena about which nothing could be done. But refinements in the science of home building have taught us the importance of maintaining inside comfort conditions regardless of outside conditions and this cannot properly be done without full consideration of the effects of the rays from the sun.

John Leonard — President,
American Society of Heating and Ventilating Engineers.

August 1, 1935.

innings the final score was Gerstenberger's team 21 runs and Berghoefer's team 17 runs. George Volk, son of J. H. Volk, a very active member of Wisconsin Chapter, was the individual star as he made four hits and scored three runs. Two very efficient umpires, Martin Erickson and Fred Krueger, kept everything in order and had complete control of the ball game at all times.

After the ball game, most of the players were ready to lie in the shade, but some had enough energy and pep to play a mighty good game of horseshoes. When the dust cleared away after about one and one-half hours, it was discovered that the champions of the horseshoe game were John S. Jung and Victor Berghoefer, who had a perfect score in the three sets played. Second place was taken by Howard Haupt and Edgar Gerstenberger. Mr. Jung and Mr. Berghoefer were dead shots on ringers and played a perfect game and they have issued a challenge to all members of the Chapter for a game at any time.

At 7 o'clock, the members gathered in the dining room of The Chalet and a very fine seven-course dinner was served. During the dinner hour prizes were awarded.

Ernest Szekely, president of the Chapter, was unable to attend the picnic and his presence was greatly missed and another missing stalwart was Prof. G. L. Larson, who has been the champion horseshoe player at all past picnics.

After the dinner and much reminiscing, the hands of the clock pointed to 9:30 and the party adjourned.

W. Roy Eichberg

Death of W. R. Eichberg

Members of the heating and ventilating industry were saddened by the untimely death of W. Roy Eichberg on July 4, 1935, at his home in Drexel Hill, Philadelphia, Pa. Mr. Eichberg was well known in Philadelphia, where he was an active member of the Society's Philadelphia Chapter, and throughout the country as a member of the national code authority for the roofing and sheet metal contracting division of the construction industry and chairman of its budget and finance committee.

Mr. Eichberg was born on October 24, 1887, at Memphis, Tenn., and for a number of years his business activities were conducted in the South. From 1905 to 1912 Mr. Eichberg was sales engineer for Beck Hammond Co., Cincinnati, O., and then he conducted the Georgia Heating Co. in Atlanta, Ga., until 1918. He was vice-president and general manager of the American Heating and Ventilating Co. in Richmond, Va., from 1918 until 1923, when he became president, and he retained this office until 1926, moving to Raleigh, N. C. From 1923 until his death, Mr. Eichberg was also president of the Carolina Sheet Metal Works, which he founded and which was located in Raleigh until 1926 when the company moved to Philadelphia.

Mr. Eichberg became a member of the American Society of Heating and Ventilating Engineers in 1929 and he had served as secretary of the Philadelphia Chapter and at the time of his death was vice-president. Mr. Eichberg was also serving his second term as president of the Sheet Metal Roofing and Heating Engineers in Philadelphia and he was secretary of the Coordinating Committee of the Construction Industry in Philadelphia.

Many members of the Society will recall Mr. Eichberg's activities as chairman of the Entertainment Committee of the Committee on Arrangements for the Semi-Annual Meeting 1934 at Buck Hill Falls, Pa. Last fall he underwent a serious operation from which he never fully recovered his normal strength and vitality. Despite his own diminished vitality, he gave untiringly of his ideas and energy to association activities and to the many problems of code administration, in addition to carrying on his own business. Even though failing in health, he retained his enthusiasm to be of service, and his sincerity of purpose, his integrity and loyalty are treasured memories of his many friends.

Mr. Eichberg is survived by his widow and two young daughters, to whom the Officers and Council extend their sincerest sympathy for the great loss that they have sustained. Also surviving are two sisters, Mrs. F. S. Wright and Mrs. G. L. Simpson, of Memphis. Mr. Eichberg's funeral was held on July 6 and was attended by many prominent members of the Society and of the Philadelphia Chapter.

John D. Small Dies

Funeral services for John D. Small, who died at his home, 411 Maple Ave., Wilmette, Ill., on July 5, 1935, after an illness of several weeks, were held at a chapel in Wilmette on July 6. Mr. Small was well known as a consulting engineer in Chicago, where he had offices at 127 N. Dearborn St.

Although a native of Jeffersonville, Ind., where he was born on December 23, 1874, Mr. Small spent all of his business career in Chicago. Starting in 1889 with the architectural firm of Burnham & Root as a draftsman, for the next 20 years he was associated with D. H. Burnham & Co., Chicago, as heating and ventilating draftsman, assistant to the chief mechanical engineer, and finally as chief mechanical engineer. Later he was heating and ventilating engineer for the National Vacuum Heating Co.

In 1913 Mr. Small opened his own office as a consulting engineer, specializing in heating and ventilating, sanitary, and power plant design.

During Mr. Small's business career, he was responsible for

John D. Small

the design of the mechanical equipment in many large projects, those in Chicago including the Marshall Field & Co. department store, First National Bank, Commonwealth Edison Building, Sisson Hotel, Surf Apartments, Aquitania Apartments, Continental Can Co.'s manufacturing plant, Chicago Tribune Tower, University of Chicago buildings, Burnham Building, Midland Building, One North LaSalle Building, Billings Hospital, W. F. Hall Printing Co.'s plant and a number of theatres, in addition to the Oliver Building, Pittsburgh, the Washington Terminal Railway Station and the John Wanamaker stores in New York and Philadelphia.

Mr. Small was elected a Member of the American Society of Heating and Ventilating Engineers in 1910 and he was active in the affairs of the Illinois Chapter, of which he was also a member. He had served as president of the *Chicago Association of Consulting Engineers.*

Surviving Mr. Small are his widow, Mrs. Pauline M. Small, a son, Stoddard Small, and a daughter, Mrs. Phyllis S. Nutt, to whom the Officers and Council have expressed their sincere sympathy, as well as several brothers and sisters.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 11 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by August 15, 1935, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
BANNON, LUCAS E., Archt., Lucas E. Bannon, Paterson, N. J.	H. F. Cox F. L. Pryor	W. H. Young, Jr. (*A.S.M.E.*) F. C. Lawton
CONNELL, HAROLD, Engr. and Estimator, Norair Engrg. Corp., Newark, N. J.	C. S. Hoffman J. F. Wait (*A.S.M.E.*)	E. A. Williams (*Non-Member*) A. F. Hinrichsen
CURRIE, F. J., Plbg. & Htg. Contractor, Kirklyn, Upper Darby, Pa.	A. E. Kriebel M. F. Blankin	M. C. Gillett H. H. Erickson
FATZ, J. L., Htg. & Vtg. Engr., Board of Education, Chicago, Ill.	John Howatt J. J. Finan	J. H. O'Brien J. F. Tobin
FRIED, H. V., Air Cond. Engr., Birmingham Elec. Co., Birmingham, Ala.	C. P. Lichty L. S. O'Bannon	E. A. Bureau (*A.I.E.E.*) W. E. Freeman (*A.I.E.E.*)
HOGUE, WILLIAM M., Sales Engr., U. S. Elec. Mfg. Co., Los Angeles, Calif.	P. C. Scofield W. E. Barnum, Jr.	J. A. Denholm, Jr. (*Non-Member*) J. F. Park
PETERSEN, STANLEY E., Air Cond. Engr., Honolulu, Hawaii.	A. H. Simonds P. C. Scofield	E. L. Cawby H. B. Edwards
SMITH, K. M., Student, Michigan State College, E. Lansing, Mich.	L. G. Miller H. B. Dirks (*A.S.M.E.*)	G. W. Hobbs (*S.A.E.*) L. N. Field (*A.S.M.E.*)
SUNDELL, SAMUEL S., Engr., Larx Co., Minneapolis, Minn. (*Advancement*).	F. B. Rowley A. B. Algren	W. F. Uhl R. E. Backstrom
TAYLOR, J. J., Partner, John Taylor & Son, Sydney, N. S. W., Australia.	J. R. Duncan Frank Bradley (*A.M.I.E.*)	B. M. Jones (*A.A.G.I.*) A. V. Hutchinson
TURNER, PRESCOTT K., Air Cond. Engr., Sawyer Lumber Co., Worcester, Mass.	H. B. Ahlberg H. H. Newell (*Non-Member*)	T. H. Morgan (*Non-Member*) J. W. Howe (*A.S.C.E.*)

Candidates Elected

In past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

FOX, JOHN H., Engr., Minneapolis-Honeywell Reg. Co., Ltd., Toronto, Ont., Can.

MACK, LUDWIG, Office Mgr., Cooling & Air Cond. Corp., Philadelphia, Pa.

MUNN, E. FITZ, Partner, Over & Munn, Winnipeg, Man., Can.

WAUNG, TSING FI, Chief Htg. & Vtg. Engr., Andersen, Meyer & Co., Shanghai, China (*Advancement*).

ASSOCIATES

ARNDT, H. W., Div. Head, Sears, Roebuck & Co., Augusta, Ga.

EMERY, GORDON W., Service Engr., H. H. Van Saun, Inc., Hackensack, N. J.

GOUEDY, K. E., Sales Engr., Modern Building Insulating Co., Atlanta, Ga.

KOCH, RICHARD G., Supervisor, Milwaukee Gas Light Co., Milwaukee, Wis.

MILLER, EDGAR R., Chief Engr., Winnipeg Cold Storage, Winnipeg, Man., Can.

MILLS, HARTZELL C., Sales Engr., Minneapolis Gas Light Co., Minneapolis, Minn.

MOON, FRANK L., Utilization Engr., Los Angeles Gas & Elec. Corp., Los Angeles, Calif.

PULLEN, ROYAL R., Asst. Master Mech., Homestake Mining Co., Lead, S. D.

SIMPSON, ARTHUR M., General Sales Mgr., Van Kannel Revolving Door Co., New York, N. Y.

WEBSTER, WILLIAM H., Jr., Field Engr., Hurst Htg. Engrs., Norfolk, Va.

JUNIORS

DENISE, JOHN R., Asst. Com'l Engr., Consolidated Gas Co., New York, N. Y.

DICK, A. V., Sales Engr., Vandecar-Harmon Co., Schenectady, N. Y.

HE CUTS THE COST of PIPING ...with a "Shield-Arc"

YOU profit with every *"Shield-Arc"* welded joint. Users claim as much as 20% to 50%. Add to these tremendous savings in initial cost, the freedom from maintenance ... for *"Shield-Arc"* welded joints are forever leakproof and trouble free. They are stronger than the pipe ... equal to the pipe in density and ductility ... and offer greater resistance to corrosion.

Even in the actual welding operation you save with a *"Shield-Arc."* It has greater uniformity of current, higher capacity, increased efficiency and eight other features which allow you to get the most out of welding ... the most in welding speed ... the most in weld quality and the most welding per dollar. Reports show the *"Shield-Arc"* saves as much as $6.50 per day due to its faster welding speed and greater efficiency.

Find out now how much better ... how much faster ... and how much cheaper you can weld piping with a *"Shield-Arc."* Ask The Lincoln Electric Company, Cleveland, Ohio. Largest Manufacturers of Arc Welding Equipment in the World. W-134

Shield-Arc welding a 6-inch steam line. Photo courtesy of The Smith & Oby Co., Power Piping Contractors and Engineers, Cleveland, Ohio.

POP "I've always believed in the 'cut and try' method. But with 'Fleetweld' electrodes, it's a case of try first and the cutting will take care of itself—in our case, the cutting was 20% of former welding costs."

LAD "Righto—and the welds made with 'Fleetweld' are equal to or better than mild rolled steel in every respect, including resistance to corrosion. Ask Lincoln for proof."

LINCOLN

"SHIELD-ARC" WELDERS

TEMPERATURE REGULATION
AIR MOVEMENT . . . HUMIDITY CONTROL
REFRIGERATING MACHINES . . . . . .
BLOWERS . . PUMPS . . . OIL BURNERS . .
STOKERS . . . . . .
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Equipment Developments

For your convenience in obtaining more information about any of this equipment, see coupon on page 95 . . . Add the new products and companies listed here to your Directory Section which you received in your January 1935 Heating, Piping and Air Conditioning and thus keep your records of sources of supply up to date throughout the year . . . Single asterisk (*) indicates equipment not listed in Directory Section; double asterisk (**) equipment and manufacturer not listed.

New Throw Away Air Filters

No. 341**—New throw-away type "Plymco" air filters have filtering medium composed of rope fibers which is light in weight, odorless, non-corrosive, and free from "splinters," according to the manufacturer. The fibers are arranged in graduated layers; from the intake side each successive layer is finer than the one preceding to distribute the impurities from the air throughout the filter according to the size of particles. Intake layer is rendered fire resistant.

The cells come in standard sizes (16x20x2, 16x25x2, 20x20x2, and 20x25x2) to fit standard filter frames, and may be had in special sizes. The filters vary in resistance from 0.08 to 0.25 W. G. with 800 cfm capacity and 300 fpm velocity, states the maker.—Plymouth Cordage Co., North Plymouth, Mass.

Electrically Operated Motor Valves for Control

No. 342—Available for steam, oil, gas, water or brine at all commercial pressures for providing close temperature control for industrial heating and cooling processes and control of heating and air conditioning in all types of buildings, new valves are actuated by oil pressure on plunger. Thermostatic switches are used to start and stop a small motor driven pump to open or close a single seated valve. Consumption is 25 watts when operating; valve returns automatically to safe position if current fails.

While used primarily for temperature control, the actuators have numerous other uses—remote push button control of valves or dampers; with electric time switches for controlling valves, dampers, or levers after given time intervals have elapsed, etc. Examples are completely automatic operation of molding presses, liquid level control, etc.—Wilbin Instrument Corp., 40 E. 34th St., New York, N. Y.

New Boiler Protector

No. 343—New boiler protector to guard against damage due to low water and the hazard from excessive boiler pressure is for use on steam or vapor systems regardless of whether the fuel be oil, gas, coal or electricity and whether the boiler be manually or automatically fired. Whenever boiler water level drops too low or boiler pressure rises too high for safety, the heat melts a fusible link in the electrode assembly of the unit. As long as this link remains intact, it holds closed the contacts of a safety switch in the main fuel supply circuit. When it melts this circuit is broken and fuel supply is cut off.

The electrode assembly and safety switch are held in a well which is surrounded by an outer housing. The water passages between well and housing are normally filled with boiler water; the housing extends into the flue passages of the boiler where it is exposed directly to the heat of the flue gases. The presence of the water in the housing keeps the well temperature low enough to prevent melting the fusible link. When the boiler water gets low enough to leave the housing, the well temperature rises and the link melts, stopping further fuel supply.—Detroit Lubricator Co., Detroit, Mich.

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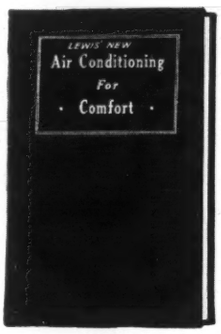

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Prepared by Samuel R. Lewis, one of the world's foremost air conditioning engineers, it embodies the most advanced knowledge of every phase of the air conditioning science. All the newest developments in systems, equipment, and practices up to June of this year are thoroughly covered. It also includes a wealth of new engineering data, gleaned from the most recent research and experience, which supersede much of the data presented in the 1932 edition. More than 75% of the material in this second edition is entirely new.

The opening chapters are devoted to the defining of air conditioning terms, and to descriptions of various types of air conditioning equipment. Then the use of the psychrometric table and chart is clearly explained, and formulas are given for every psychrometric computation covering both humidifying and dehumidifying. Succeeding chapters take up the various functions of the air conditioning system, and show exactly how each is accomplished. Next, the various factors, such as heat transmission, infiltration, etc., which must be taken into account in planning any system, are described in detail. Then—employing the data presented in 13 earlier chapters for all calculations—it outlines for the reader in full detail the correct design of a complete heating and cooling system, first, for a specific residence, and then for a large building housing different activities.

Engineer, contractor, manufacturer, salesman, instructor, and student alike can add immeasurably to their knowledge of every aspect of air conditioning from this most understandably written and newest of all air conditioning books. Mailing of the coupon below, with remittance of $2.50, will bring a copy to you promptly. Your money will be refunded on return of the book if it does not prove satisfactory in every way.

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Compiled principally from material published originally in HEATING, PIPING and AIR CONDITIONING, this volume embodies the accumulated practical knowledge of 25 outstanding air conditioning specialists. Every phase of the whole subject of industrial air conditioning—from fundamentals and equipment, through the design, installation, operation, and maintenance of the different types of systems, and the problems relating to each—is explained thoroughly and plainly by these men from the standpoint of their successful experience.

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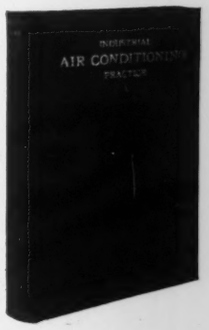

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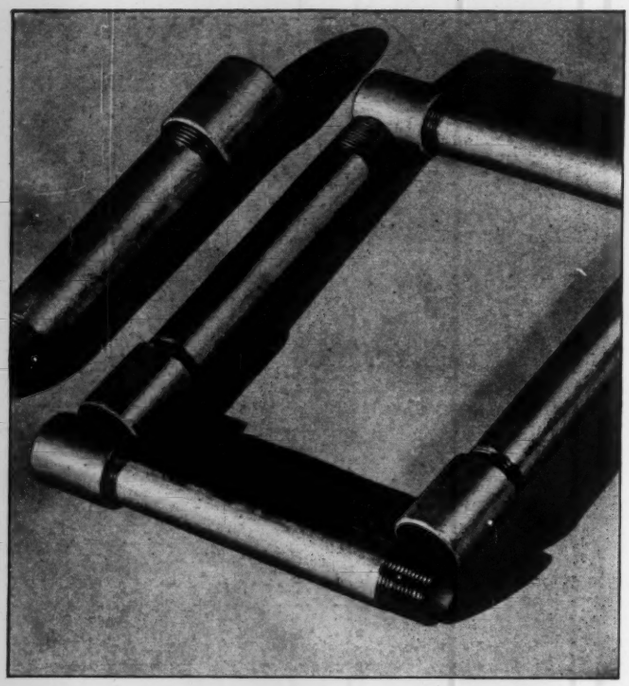

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Motor Operated Shielded Psychrometer

No. 344—Elimination of error due to heat radiation from surrounding objects, including the operator, is achieved by a new electric thermo-shield psychrometer. The air sample is drawn through the top, away from proximity with the operator. The initial warmth effect of wet bulb reservoirs is eliminated by individual applications of water to the wet bulb for each reading, states the maker.

The entire mechanism is encased in a highly polished chromium plated thermo-shield equipped with a protective glass window for thermometer visibility and an integral rack for psychrometric tables. Thermometers are 10 in. long, carefully matched, and graduated minus 5 to plus 130 F in ½ deg increments. The fan is operated by a 110 volt a-c motor.

The case is approximately 7 in. square by 13 in. high, and is provided with a carrying handle, and sufficient electrical connector to assure portability and flexible operation.—Julien P. Friez & Sons, Inc., 4 N. Central Ave., Baltimore, Md.

Adjustable Flow Meter Orifice

No. 345—An adjustable orifice which automatically shifts its position in accordance with major changes in rate of flow when installed in a pipe line and connected to a standard recording and integrating fluid meter, widens the flow range which may be accurately covered by the meter.

The orifice is similar in appearance to a hydraulically operated gate valve; the gate, however, is a sharp edged orifice segment, coefficients for which have been accurately determined. Its position varies the size of the orifice opening through which the fluid measured must pass.

Under an increasing flow condition, the orifice shifts position at a higher rate of flow than it does under a decreasing flow condition. This overlap prevents constant shifting when conditions are such that the flow holds closely to the changeover point. —Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland, Ohio.

New Small Motor Starter

No. 346—Pushbutton motor starter combining new simple mechanism and modern appearance is adaptable for almost any fractional horsepower application. Switch provides protection against overloads with a free-tripping thermal overload mechanism. An overload is indicated in the switch itself by the return of the operating button to the "off" position; pushing the "start" button resets the overload mechanism and restarts the motor.

Capacity can be varied by changing the heater coil, which is readily accessible by removing cover plate and two screws. Other features include solid silver to silver "twin-break" contacts, mechanism removable for wiring, accessible terminals, non-rusting mechanism, stainless steel springs, and top and bottom knockouts.—Cutler-Hammer, Inc., 421 N. 12th St., Milwaukee, Wis.

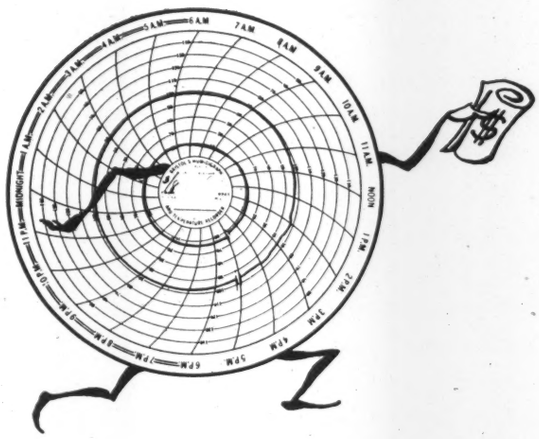
Sheet Metal Shears

No. 347—New line of squaring shears with gap in the housings and for foot operation recently announced. Gap permits cutting sheets longer than the cutting length of the shear. Shears are made in five lengths for cutting sheet metal up to 16 gage in thickness. Among the features, says the maker, are ease of operation, portability, rigid construction, accurate cutting, complete accessibility at the back of the machine for easy removal of sheared pieces and for setting gages.—Niagara Machine & Tool Works, 637 Northland Ave., Buffalo, N. Y.

Condensing Units, ¼ to 20 Hp

No. 348—New commercial condensing unit line consists of 128 units to meet all temperature requirements for commercial refrigeration and air conditioning. Ten standard blocks, one for each motor size, provide for a range from ¼ to 20 hp.

Cylinder blocks, pistons, valve plates, cylinder heads and com-



CHARTS *bring home the cash!*

Contractors who sell humidifying, air conditioning or heating equipment, often encounter buyers who are dissatisfied with the equipment they have just installed. It is not making good. It is not meeting its guarantees. And so on.

In such a situation the one thing *not* to do is to argue. Facts, and facts alone, are needed. Install a Bristol's Thermo-Humidigraph for a few days and from its 24 hour or 7 day record you get a continuous uninterrupted chart of the temperature. And of the relative humidity. Every fluctuation is clearly revealed—the time it occurred and its duration.

As one contractor, Mr. A. H. Angermeyer, Neenah, Wis., recently put it, "These Bristol's charts told the story. We were able to make proper adjustments, and we received our cash."

Right: Bristol's Portable Thermo-Humidigraph or combination relative humidity and temperature recorder, Model 4069. Eight inch chart, one revolution in 24 hours. For permanent wall mounting, specify Model 4068.

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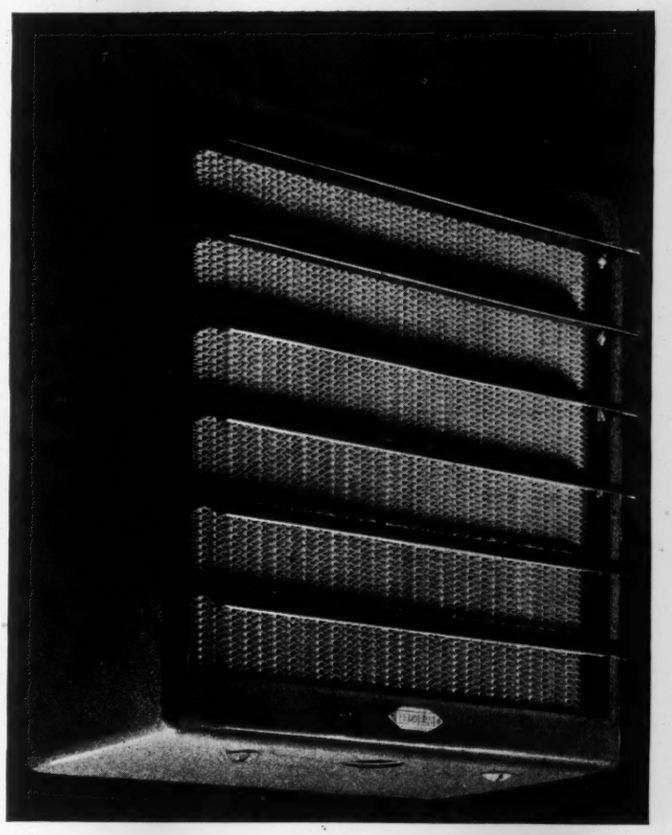

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Streamline tubes, patented fin construction, graduated pitch fans, resilient motor mountings assure quieter operation.

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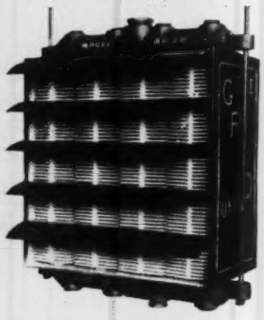

In it there are no two dissimilar metals to cause electrolytic action; hence no corrosion. The Grid Unit Heater "fins," or heating, sections, are special alloy aluminum cast, metal to metal contact with the steam chamber—easy to clean—and due to the design of the patented "fin" sections provides more air changes per hour, delivering heat direct to the working zone—no maintenance expense—sturdy construction throughout, built to last a long time.

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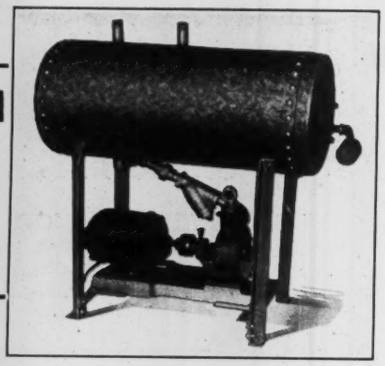

Where the city water pressure is lower than the desired steam pressure, the M-K-O Boiler Feed Pump will automatically maintain a constant water level and supply make-up water as required. The unit is rugged, efficient and compact, and while designed for use with the Kane or Ofeldt Boilers, it can be used with any other as well.

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pressor housings are made of a fine grain, cast iron solution, specially formulated for refrigeration service, states the maker. Cylinder walls, piston pin holes and bearings are diamond bored for efficiency and quiet operation. By placing both the suction and discharge valves in the cylinder head, the gas refrigerant does not enter the crankcase and oil pumping is eliminated. The valves are of the flat, reed-type, made of high-grade Swedish steel.

Large condensers with increased capacities, heavy bases, improved appearance, quiet operation and more flexibility are provided in the units, which are available for sulphur dioxide, methyl chloride and freon.—Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich.

Induced Draft Cooling Towers

No. 349—Induced draft cooling towers have capacity ranging from 5 to 200 gpm. These towers are recommended by the maker for refrigeration plants having from 1 to 50 tons refrigeration capacity and for industrial plants using Diesel engines and other internal combustion engines which need jacket water cooling for effective operation.

Water distribution is secured through low pressure clogproof "Rotojet" nozzles, which spray a finely atomized mist of water downward through upward currents of air induced by the fan assemblies placed at the top of the tower to assure maximum water and air contact. During cool weather, where towers are required for year around operation, the fans may be shut off and ample cooling capacity secured without their use. Depending upon tower size and capacity, one or more fans are used, the fans being driven by individual splash and weather proof motors at speeds of approximately 450 rpm. All towers of the type "K" series are of heavy steel frame surrounded by a sectional copper-bearing galvanized steel housing asphalted and aluminum painted inside and out before shipment. Due to standardized features, all units can be quickly assembled on the job by ordinary labor without skilled supervision.—Binks Mfg. Co., 3113-50 Carroll Ave., Chicago, Ill.

General Duty Welding Torch

No. 350—New welding torch ("Purox No. 35 General Duty") has extruded brass handle of ribbed design which reinforces the torch, enables it to be easily gripped. Connections are ferrule type and extend parallel to the axis of the handle. Individual mixers in each welding head are designed for maximum resistance to flashback; at the same time the effective range of each size tip is greatly increased, says the maker. Monel metal seating surfaces are easily removable for cleaning or replacement and tips are of hard drawn copper.—Linde Air Products Co., 205 E. 42nd St., New York, N. Y.

Simplified Group Filter Frame

No. 351—New type of group filter frame for "Airplex" dry type air filters is built without any cross-pieces, the filters furnishing their own seal and support. The group frame is built up and shipped out integrally in sizes holding from two to twenty-five cells; when more units are required, two or more frames are bolted together.

Filter cell is fed into top of frame and pushed down in the U shaped frames, the last one being held by a simple compression catch. Each cell has one layer of cotton wadding on bottom edge to form an air-tight joint. New bulletin issued by manufacturer describes these filters and the new frame and gives application and performance data.—Davies Air Filter Corp., 390 Fourth Ave., New York, N. Y.

Propeller Type Comfort Coolers

No. 352—New line of suspended propeller type comfort coolers for chilled water or direct expansion is designed for commercial and the smaller industrial air conditioning applications. The four direct expansion units have capacities of 870, 920, 1720,

and 3200 cfm of standard air, and Btu capacities range from 1,940 to 22,170; 2,760 to 31,600; 4,220 to 48,400; and 7,560 to 87,000, respectively depending upon difference between entering air dry bulb and refrigerant temperature and difference between entering air dew point and entering refrigerant. Corresponding capacities for the water or brine types are 2,560 to 25,800; 3,470 to 34,900; 5,320 to 53,500; and 9,420 to 95,000.

Units are of draw through type, the fan pulling the air through the coil and discharging through a "Freeflo" grille.—The Trane Co., LaCrosse, Wis.

New Valves for Copper Pipe

No. 353—New valves for use with Type M Mueller Brass Co. copper piping are known as stream-line or soldered joint end valves, can readily be soldered to the pipe. They are available in globe and angle, rising stem and non-rising stem gate, and swing check valves.

The joint is made by feeding wire solder through a feed hole in the valve after the copper pipe has been assembled in it, after which heat is applied by a blow torch. The liquefied solder is carried around the surface between pipe and valve by capillary attraction and becomes visible at the outer edge between the pipe and valve.—The Wm. Powell Co., Cincinnati, Ohio.

Reducing Duct Elbow Friction

No. 354—Designed to reduce friction losses in duct elbows enabling use of right angle turns instead of long radius elbows, "Ducturns" (installed behind the grille or register opening) are composed of tubular blades shaped to turn high velocity air with minimum friction loss.

All points on the periphery of one blade are equidistant from similar points on an adjacent blade, states the manufacturer, to prevent energy losses through partial conversion of velocity to static and back to velocity pressure.—Tuttle & Bailey, Inc., New Britain, Conn.

In the **article by Louis A. Harding** published in the July, 1935, H. P. & A. C., pp. 317-322, the first line of the last paragraph, first column, p. 322, should read "five tube row" instead of "three tube row." Also, in the second column, p. 318, the equation should read:

$$0.24 \times 135\ (80 - t_3) = 220.$$

$t_3 = 73.2$ F with a differential of 6.8 F instead of 8 F as stated.

Book Reviews

Lewis' New "Air Conditioning for Comfort"

"Air Conditioning for Comfort," by Samuel R. Lewis, Consulting Engineer. 2nd edition, 1935. 277 + vii pp. and large psychrometric chart, 5¼x8¼ in., clothbound. Published by Keeney Publishing Company, 6 N. Michigan Ave., Chicago, Ill. $2.50.

This new edition of "Air Conditioning for Comfort," by Samuel R. Lewis, has been completely re-written and enlarged. The record sheets for the computation of the heating and cooling loads for residences and commercial buildings are new and have been simplified. Two chapters not included in the first edition explain in detail the design of a year 'round air conditioning system for an office building, which includes banking space, offices, stores, restaurant, cocktail lounge, and kitchen. The necessary computations are given in their entirety in the explanation of the design procedure. Similarly, two re-written chapters cover the design of a year 'round residence air conditioning system, with complete computations, record sheets to simplify the various steps, and the necessary tables and charts.

Chapter 1 defines the terms used in air conditioning, and in Chapter 2 types of air conditioning systems and equipment are

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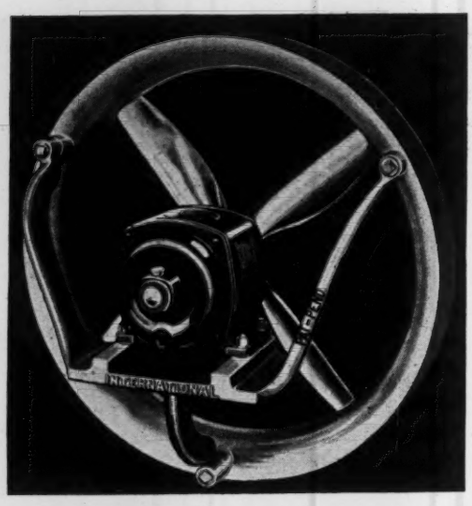
Water is also the enemy of every underground conduit system. Before you select *any* conduit for your underground steam pipe jobs, be sure it has an *adequate external underdrain* to carry off water and keep the insulation dry and the lines efficient. ●

On a recent multiple pipe industrial job, installation labor cost on standard 18" type F conduit was 47c per foot.



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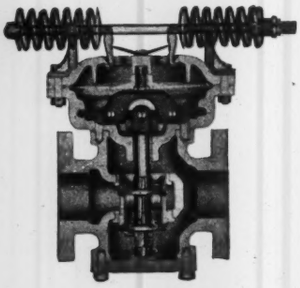

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AUTOMATIC VALVES FOSTER EST. 1879 · ENGINEERING CO.

for STEAM, WATER AND AIR

This is the Foster Type W Regulator which is widely used on steam, water and air lines for initial pressures up to 250 pounds and with delivery pressures ranging from 10 to 125 pounds. It is recommended for general but not for deadend service. It is direct-acting with pressure changes acting directly on a large diaphragm. There is no "stuffing box." Sizes ½" to 20". Also made with enclosed direct-acting spring.

Foster makes a complete line of Automatic Valves for practically all classes of service involving the control of steam, liquids and gases.

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described. Chapter 3 explains the psychrometric chart and table and its use, and in Chapter 4 formulas and methods of computation are presented; in this chapter eighteen different computations encountered in air conditioning design work are explained and examples are given. Chapter 5 takes up humidifying and dehumidifying, with descriptions of methods and equipment which include numerous typical examples. In Chapter 6, refrigeration for air-conditioning (including water vapor) is covered.

Chapter 7 discusses air conditioning and the human body, comfort and health, with data and curves based on the work of the *American Society of Heating and Ventilating Engineers*, including the comfort chart. Chapter 8, on heat transmission through buildings, includes tables of design temperatures for both heating and cooling, and ten full-page plates of coefficients of heat transmission through various types of building construction.

In Chapter 9, information and tables on heat from the sun and appliances is given and the effect of heat gain from these sources on the cooling load is discussed. Chapter 10 considers design air volumes, infiltration and ventilation and Chapter 11 is devoted to air distribution within rooms and duct design and layout; this chapter (one of the most important, as the success of an air conditioning installation depends upon the air distribution) includes many tables and examples. Chapter 12 covers water circulation in heating and cooling systems, with examples of the design of piping for air conditioning installations.

Chapter 13, a new one, explains the estimation of operating costs and includes a curve of operating hours for cooling plants based on design wet bulb temperatures. The last four chapters of the book (Chapters 14-17) are devoted to the detailed design of year 'round air conditioning systems for a residence, and for a commercial building, as mentioned.

Booklets, Reports and Papers

"Review of Literature on Effects of Breathing Dusts, with Special Reference to Silicosis," by D. Harrington and Sara J. Davenport. Information Circular 6835, *Bureau of Mines, Washington, D. C.* 68 *pp.*, 8x10½ *in., mimeographed.*

This circular is Part 1 of a review of the literature on effects of breathing dusts, with special reference to silicosis, largely in the mining and allied industries; Parts 2 and 3 will deal with prevention and treatment of dust disease., and economic and legal aspects.

Part 1 comprises chapters on definitions and classification of dusts, sources of exposure, and physiological effects. It includes a bibliography of 145 references.

Conventions and Expositions

American Welding Society: Fall meeting, Chicago, Ill. Sept. 30. Secretary, W. Spraragen, 29 W. 39th St., New York, N. Y.

Midwest Power Show: October 14-18, Chicago, Ill. G. E. Pfisterer, Managing Director, 1 N. LaSalle St., Chicago, Ill.

American Gas Association: Annual convention, October 14-18, Atlantic City, N. J.

American Public Health Association: Annual meeting, Oct. 7-10, Milwaukee, Wis. Headquarters Office, 50 W. 50th St., New York, N. Y.

International Acetylene Association: Annual convention, Nov. 12-15, Cleveland Hotel, Cleveland, Ohio. Secretary, H. F. Reinhard, 30 E. 42nd St., New York, N. Y.

International Heating & Ventilating Exposition: January 27-31, International Amphitheater, Chicago, Ill. Manager, Charles F. Roth, Grand Central Palace, New York, N. Y.

American Society of Heating and Ventilating Engineers: Annual meeting, January 27-31, Palmer House Hotel, Chicago, Ill. Secretary, A. V. Hutchinson, 51 Madison Ave., New York, N. Y.

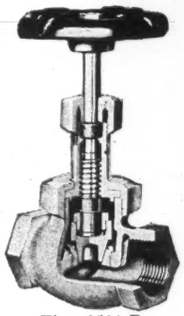

Fig. 0501-P Renewable Plug and Seat

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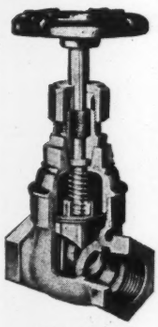

Fig. 0201 Renewable Seat Rings

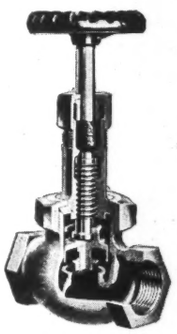

Fig. U-01 Renewable Composition Disc

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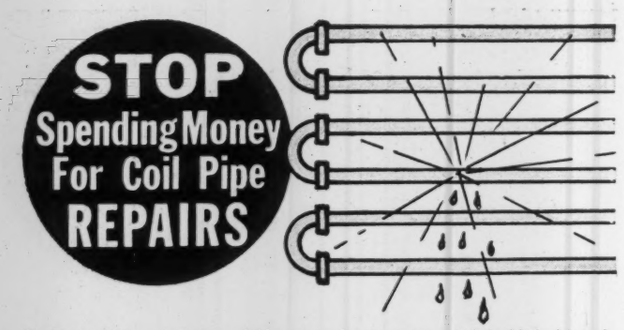

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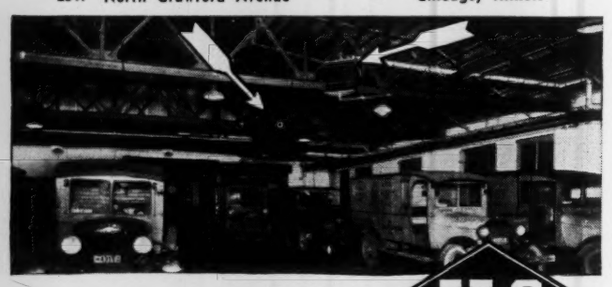

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No. 512. AIR CONDITIONING: Frick Co., W. Main St., Waynesboro, Pa. 16 page booklet on air conditioning commercial and industrial buildings, residences and miscellaneous applications, picturing numerous installations. Ammonia, carbon dioxide, and freon refrigerating equipment is described as are low pressure refrigerating units charged with methyl chloride or freon.

No. 513. AIR CONDITIONING: York Ice Machinery Corp., York, Pa. Looseleaf air conditioning sales manual, the first section devoted to brief descriptions and reports of results for a number of selected installations, and the second section presenting data on the need for air conditioning, definitions of terms, the three markets for air conditioning (profit, comfort and health, and industrial processing), specific applications and advantages, and approximate constants for giving a preliminary idea of the required capacity, initial and operating cost of a particular installation. The chapter on specific applications and advantages of air conditioning is particularly complete and helpful, and the data on approximate constants should prove most useful.

No. 514. AIR CONDITIONING: York Ice Machinery Corp., York, Pa. Series of bulletins on air conditioners and refrigerating units, including 10, 15, 20 and 25 hp freon condensing units and horizontal air conditioning units for year 'round services.

No. 515. AIR FILTERS: W. R. Ripley Co., Tacoma, Wash. Bulletins on "Relay" and "Edgeseal" filters describing construction, features and giving technical data and ratings.

No. 516. COMBUSTION CONTROL: Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland, Ohio. 32 page bulletin describing control system for automatically maintained steam pressure, combustion efficiency and furnace draft, describing principle of operation, applications and the equipment for use with boilers from 200 hp up, fired either with fuels in suspension or with stokers.

No. 517. DIAPHRAGM VALVES: C. J. Tagliabue Mfg. Co., Park & Nostrand Aves., Brooklyn, N. Y. 16 page booklet on diaphragm valves, giving description of construction of various types, rating curves, and steam flow charts.

No. 518. DIESEL ENGINES: The Cooper-Bessemer Corp., Mt. Vernon, Ohio. 12 page booklet describing in detail single and twin cylinder horizontal diesel engine, 25-110 bhp, pointing out industrial applications, describing operation and construction, and giving rating curves and general data.

No. 519. DRIVES: Rockwood Mfg. Co., 1801-2001 English Ave., Indianapolis, Ind. Four page bulletin describing several applications of short center flat belt drives, including their applications to compressors.

No. 520. DUST COLLECTORS: Pangborn Corp., Pangborn Blvd., Hagerstown, Md. 24 page bulletin describing and picturing details of all metal cloth screen dust collectors including features, screen rapping mechanism, standard sizes and dimensions, specifications. Information on types of dust collectors and auxiliary equipment is included.

No. 521. HEAT TRANSFER SURFACE: Young Radiator Co., Racine, Wis. 20 page catalog of heat transfer surfaces for air conditioning and commercial applications, describing complete

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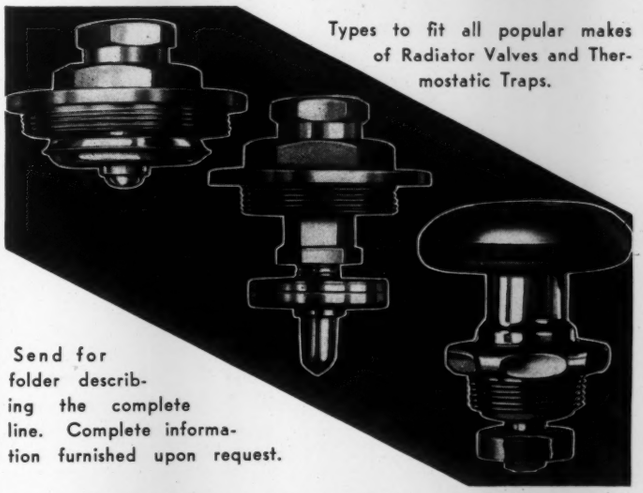

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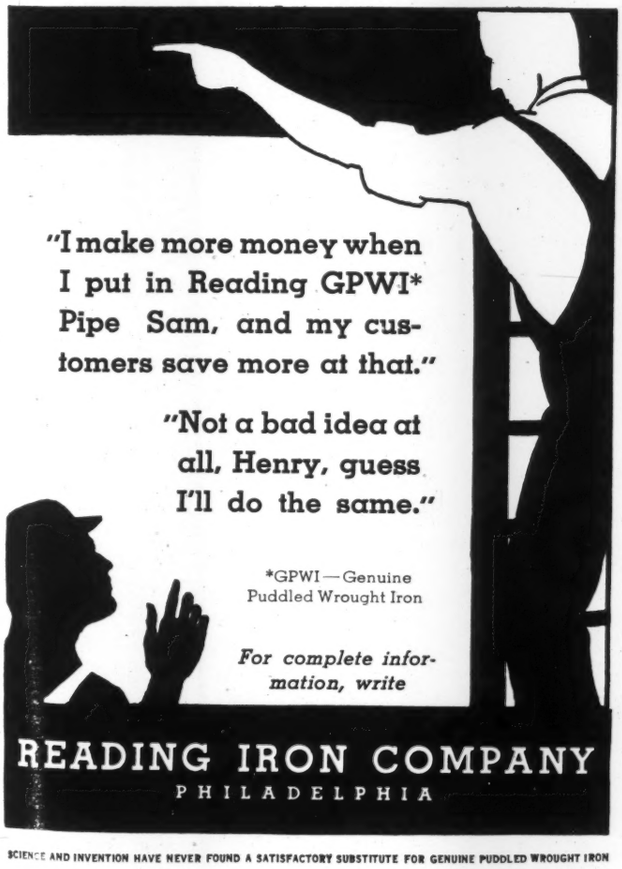
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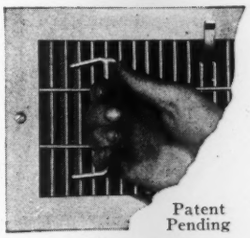

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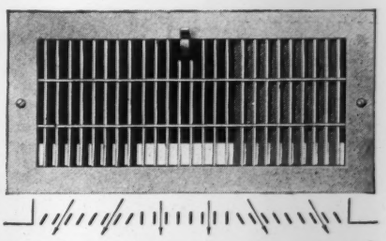

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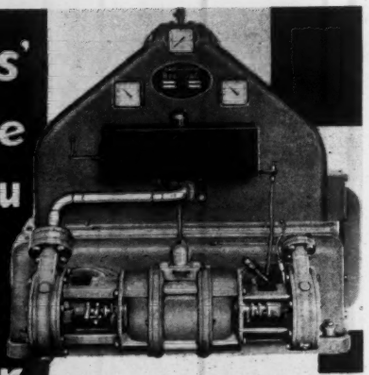

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No. 530. RADIATORS: Richmond Radiator Co., Inc., 1480 Broadway, New York, N. Y. Leaflet describing cast iron convector type "Richvar" radiators for steam, vapor, vacuum or hot water heating, describing construction and giving ratings.

No. 531. REGULATORS: The McAlear Mfg. Co., 1901-09 S. Western Ave., Chicago, Ill. 96 page bulletin describing regulators for reducing and controlling steam, water, air, gas and oil pressure, including engineering data and charts and blueprints showing practical applications and roughing-in dimensions.

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No. 533. SURFACE CONDENSERS: Worthington Pump and Machinery Corp., Harrison, N. J. Four page bulletin de-

scribing and illustrating surface condensers of welded steel shell construction with folded tube layer arrangement.

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No. 535. VALVES: The Fairbanks Co., 396 Lafayette St., New York City. 56 page two-color catalog giving descriptions, illustrations and prices for valves and unions with complete information about each type and its features.

No. 536. WELDED PIPING: Air Reduction Sales Co., Lincoln Bldg., 42nd St. opposite Grand Central, New York, N. Y. 24 page booklet, "Facts About Welded Piping" addressed to heating engineers and architects and of interest also to building and plant executives responsible for piping installation and maintenance. Numerous illustrations supplement the text in describing the advantages of welded piping. Welding fittings are briefly described, information is given on brazing, and a special section gives pipe welding specifications for heating, plumbing, refrigeration and power piping installations.

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